POWER PLANT TESTING

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A MANUAL OF TESTING ENGINES, TURBINES, BOILERS, PUMPS, REFRIGERATING MACHINERY, FANS, FUELS, MATERIALS OF CONSTRUCTION, ETC.

BY

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PREFACE

In the preparation of this book the object in view has been primarily to give in a small volume, somewhat in detail, the generally approved methods of testing engines, turbines, boilers and the auxiliary machinery usually found in power plants, as well as to present more or less complete descriptions of the various kinds of apparatus used and the calibrations required for accurate testing. In addition to this subject-matter, chapters have been prepared on the testing of fuels, refrigerating and hydraulic machinery, as well as on the proper methods and the machinery to be used in making tests of the strength of the materials commonly used in the construction of buildings.

As a book for students in laboratory courses it is intended particularly for use in large classes in which at the beginning of the laboratory periods it is necessary to begin at the same time a number of different experiments and tests. On this account care has been taken to state as clearly as possible the descriptions of the apparatus to be used and the precautions to be observed to secure accuracy in the results. Students should be expected, however, to rely to some extent on their own initiative.

In most respects the book is probably complete enough in descriptive matter and in general instructions so that very little lecture-room work is needed for at least elementary courses. It is the author's opinion that students in experimental engineering laboratories should not receive a great deal of assistance in planning and conducting tests. Sometime they must learn to be resourceful and independent of the "school" type of instruction and obviously the sooner this is appreciated by both instructors and students the greater will be the benefits. At least for very small classes the better plan is the one advocated years ago by

a famous educator, that students working in laboratories when assigned the work of testing a machine, a new type of aeroplane engine, for example, should have very simple instructions such as: "Make tests of this new type of engine, find out what you can about it and report your results." It is to be hoped that the particular method of teaching in laboratories known familiarly as "feeding with a spoon" has disappeared in present-day instruction in technical schools and colleges.

Quite a large part of the training required for one to become accurate and reliable in the work of observing and interpreting the results of tests of machinery consists in becoming familiar with the details of the adjustment and calibration of the various instruments, so that they may be used intelligently.

Although in the arrangement of the chapters the use of the book by students was given the most careful consideration, yet as a whole the needs of the "practical" man were not lost sight of, and it is hoped that the author's experience when working with this group of readers in testing both large and small power plants has helped to make the book interesting and helpful to them. The book is intended to be also a manual giving useful information in a more or less limited way to those professional engineers having the advantages of a technical training, but who are not thoroughly familiar with the most up-to-date methods of testing.

In many cases not nearly all the results that should be calculated to make up a complete report are mentioned. It is the opinion of the author that in a text-book it is desirable that no more than very general instructions should be given regarding the conduct of the test, the quantities to be calculated, and the form and tabulations expected in a report. Such details should be left in the hands of the instructor. Because the size of the book was limited it was necessary to omit explanations of the methods of calculating many interesting and more or less applicable results from tests. In the more extended courses it is believed that the instructors can readily fill in these omissions.

The author is particularly indebted in the preparation of this book to Professors M. E. Cooley, J. R. Allen, C. J. Tilden, and E. D. Campbell, of the University of Michigan; Professors I. N. Hollis and L. S. Marks, of Harvard University; Professor H. W. Spangler, of the University of Pennsylvania; Professor W. F. M.

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The short steam tables given in the Appendix have been taken, with permission, from Allen and Bursley's "Heat Engines." In many cases, particularly regarding engineering practice abroad, Pullen's "Engines and Boilers" has been found very useful.

J. A. Moyer.

Ann Arbor, Michigan,
August, 1911.

INTRODUCTION

TESTS of the machinery in a power plant are usually made to determine the capacity and efficiencies of its various units when operating under certain definite conditions. In recent engineering practice manufacturers and contractors are generally required to make certain estimates and guarantees of the capacity and efficiency of the various kinds of machinery supplied. This is exactly equivalent, in other words, to agreeing to provide for doing a given unit of work under specified conditions at a definite cost. The purchaser, on the other hand, for his protection, finds it necessary to determine from the results of reliable tests whether the "guarantees" can be obtained. Obviously, then, the importance of knowing how to make careful and reliable tests, of which the results will not be questioned, cannot be overestimated.

Tests of power plants as a whole are also necessary and should be made from time to time in order to determine what results can be obtained from an economic viewpoint when operating under the existing conditions; and also for finding out what saving can be obtained by changes in the operating conditions or by the installation of more efficient auxiliary machinery. From the viewpoint of determining whether or not it is economical to replace old equipment with new, tests of old installations are relatively more important than those of newer ones, because usually it is out of the question to relegate practically new machinery to the scrap-heap. The greatest importance of such tests is due, however, to the fact that they show how nearly the existing conditions of operation conform to those of standard engineering practice, and to those obtained in other plants operating with the greatest success.

Practice tests in the laboratory are intended to show to students by actual experience the best methods for investigating the problems arising in the operation of plants, how to work out in a practical way the doubtful points in designing and constructing machinery, and, above all, to think accurately and systematically in such matters.

Procedure for making accurate tests may be stated as follows:

1. Procuring a suitable standard testing equipment. Any instruments and apparatus not well known to engineers generally and which are of doubtful accuracy or sensitiveness should always be avoided. Remember that a single element of uncertainty may vitiate the acceptance of the results of a test of otherwise undoubted accuracy.

2. Careful calibrating of instruments before a test, so that the greatest possible errors of the tests are definitely known and that proper allowance can be made in the results.

3. Systematic recording of observations.

4. Recalibrating of instruments after a test to determine whether there have been any changes in their accuracy.

5. Preparing of a report embodying data, results and conclusions.

6. Tabulating and plotting on cross-section paper the important results. This plotting is not only for the purpose of showing the results graphically, but also for the purpose of providing a check or a method of eliminating errors in observations or in calculations. The skill of an engineer in testing is shown more than in any other way, by his **ability to check results.** If after applying various checks, usually by means of plotted curves, the results for varying conditions are found to agree, the engineer is able to tell definitely whether or not his tests are reliable.

CONTENTS

POWER PLANT TESTING

CHAPTER I

MEASUREMENT OF PRESSURE

THE simplest instrument used for measuring pressure is a glass tube bent into the shape of the letter U, as illustrated in **Fig. 1.** When such a tube, called technically a **manometer** or **U-tube,** is partly filled with a liquid, usually water or mercury, and is connected at **A** by means of tubing to the vessel in which the pressure is desired, there will be observed a difference in the level of the liquid corresponding to the pressure. If the end of the tube at **B** is open to the atmosphere, then the difference in the level of the liquid in the two legs measured in inches, multiplied by the weight of a cubic inch of the liquid in pounds, gives the difference in pressure in pounds per square inch between that in the vessel and atmospheric pressure. When the level in the leg **B** is higher than in **A** then the pressure measured is greater than atmospheric and is called **gage pressure** to distinguish it from the other condition when the level in the leg **A** is higher than in **B**; that is, when the pressure is less than atmospheric. In the latter case we speak of **vacuum** or negative pressure.

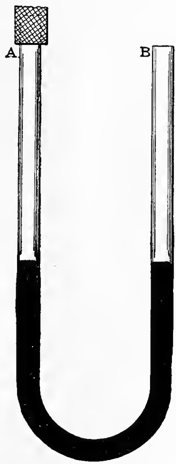


FIG. 1.—A U-tube. The Simplest Instrument for Measuring Pressures.

As such instruments are usually constructed, a scale suitably

graduated for measuring the difference between the levels of the liquid in the tube is placed between the two legs, as shown in **Fig. 2.** Still another type is illustrated in **Fig. 3.** In a manometer of this kind one leg can be made very short if it is correspondingly large in diameter. If the scale is adjusted so that the level in the short leg is at the zero of the scale, then the level in the long leg will indicate directly inches of pressure or of vacuum as the case may be. A typical vacuum gage of the same kind is illustrated in **Fig. 4.** The end of the tube corresponding to the short leg in **Fig. 3** is shown at **A.** When manometers are to be used for pressure or vacuum measurements of **steam,** a condenser (**C, Fig. 5**) is often employed to prevent the passage of steam into the glass tube in which it would form a water column on the top of the mercury for which a correction[1] would have to be made. To be effective the condenser, **C,** must always be partly filled with water. Manometers or U-tubes of very small diameter when filled with mercury may be affected by capillarity to such an extent that in order to obtain the true height corresponding to the pressure, a correction must be added. It is not at all unusual to find manometers used for vacuum gages to be comparatively small in diameter, and unless the graduations of the scale have been corrected for the error due to capillarity the proper allowances must be made for all observations. **Fig. 5** shows by a curve the values of this correction as determined by Pullen for mercury columns of various diameters.

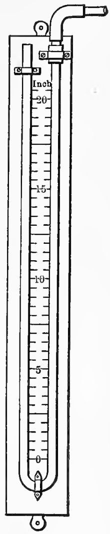


Fig. 2.—A Simple Manometer with a Graduated Scale.

Mercury columns should be read at the top of the meniscus and water columns should be read at the bottom. In this way,

[1] Correction for water on the top of a mercury column is most conveniently made by dividing the length of the water column by the specific gravity of mercury (13.6) and adding this equivalent length to the mercury column on which the water rests.

except in very small tubes, the errors due to capillarity may be regarded as negligible.

Conversion of Pressures. It is frequently necessary to reduce pressures in inches of mercury or of water to the equivalent in pounds per square inch. Since the weight of a cubic inch of mercury at 70 degrees Fahrenheit is .4906 pound and

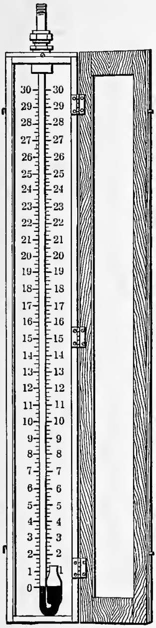


FIG. 3.

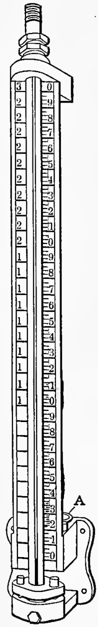


FIG. 4.

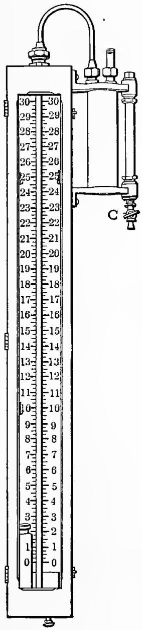


FIG. 5.

Typical Mercury Vacuum Gages.

of water at the same temperature is .0360 pound, pressures in inches of mercury at the usual "room" temperatures can be reduced to pounds per square inch by multiplying by .491 or by dividing by 2.035, and similarly inches of water can be converted to pounds per square inch by multiplying by .0360 or by dividing by 27.78.

Kilograms per square centimeter are reduced to pounds per square inch by multiplying the kilograms per square centimeter by 14.223, or by dividing by .0703.

A cubic foot of water at 70 degrees Fahrenheit weighs 62.3 pounds and at 30 degrees Fahrenheit, 62.4 pounds. At ordinary room temperature the pressure due to 2.31 feet of water is equivalent to one pound per square inch.[1]

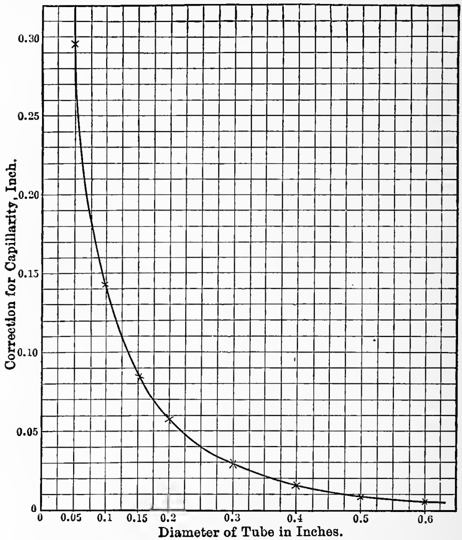


FIG. 6.—Curve of Capillarity Corrections for Mercury Columns.

Tubes used as mercury manometers must be cleaned from time to time by washing the inside surface with nitric acid and afterward thoroughly cleansing them with water. Mercury used in manometers should be free from impurities. Usual

[1] The unit pressure of one pound per square inch is equivalent also to that due to a column of air of uniform density, of which the vertical height in feet is approximately, 144.0 divided by the weight of a cubic foot of air at the temperature, pressure and humidity as observed. Tables of the weight of air are given on page

impurities can generally be removed by filtering through a clean cloth of close texture or a thin chamois leather. Air can be removed by boiling, but by far the best method for cleaning mercury is by means of a mercury still. Unfortunately an apparatus of this kind is not available in most engineering laboratories.

Pressure Gages. The large size necessary, however, for manometers or U-tubes, even if filled with the heaviest liquids, makes their use unsuitable except for comparatively low pressures. Instruments more desirable for high pressures are made by the application of some kind of elastic material designed to produce a uniform deformation for variations of pressure. By connecting a suitable auxiliary mechanism to the elastic element it can be made to move a needle to indicate on a graduated dial the degree of pressure. The most common form of such devices is a hollow brass or steel tube bent into the shape of an arc of a circle. It is a well-known principle that when a straight piece of tubing is bent into this shape the sides come nearer together, making the section of the tube a very much flattened oval. A tube of this kind is illustrated in **Fig. 7,** showing also in the right-hand corner a transverse section. If one end of such a tube is closed and fluid pressure is applied to the inside, the parallel sides, as at **A** and **B,** tend to separate and consequently there is a tendency for the radius of curvature of the tube to become larger, thus moving the end at **E** toward **F.** By connecting a suitable mechanism

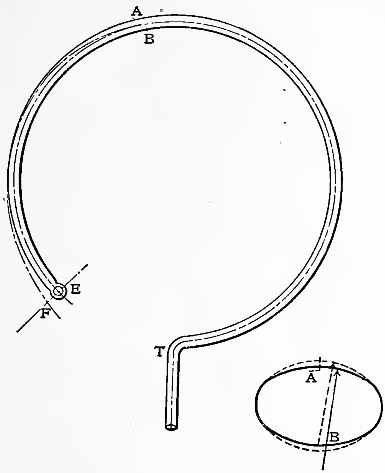


Fig. 7.—A Typical Bourdon Tube.

to **E**, the degree of pressure can be indicated. Instruments of this kind are called **Bourdon gages.**

Fig. 8 shows one of the simplest forms of such gages used in power plants to indicate the pressures. It consists essentially of the curved tube **T** of oval cross-section closed at one end. By means of suitable levers and gears a pointer or needle **P** is made to move over a dial graduated or marked to indicate pressures in standard units as, for example, pounds per square in. (English system) or kilograms per square centimeter (Metric system). **(Fig. 9.)**

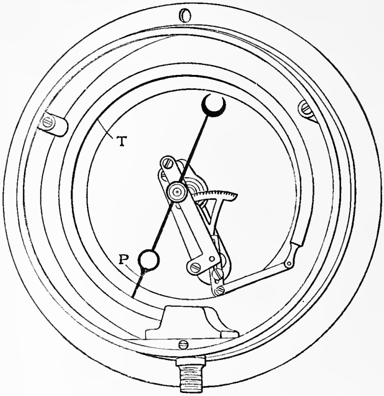


Fig. 8.—Typical Bourdon Pressure Gage.

Fig. 10 shows a form of Bourdon gage in which the amount of vibration of the needle due to the jarring that occurs in

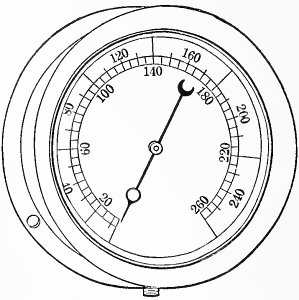


Fig. 9.—The Dial of a Pressure Gage.

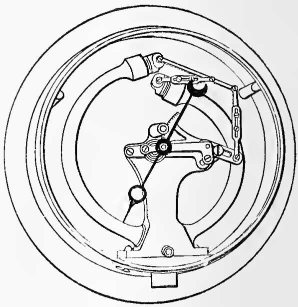


Fig. 10.—A Modified Bourdon Gage.

locomotive and other portable services has been reduced to a minimum by supporting the pressure tube in the middle

instead of at its end as in **Fig. 8.** This form of tube has also advantages for use in gages exposed to temperatures below freezing, since the arms can be drained of water, while the other form will usually hold the water that has entered.

Gages to be used to determine the pressure of ammonia have the oval tube made of steel instead of brass because the latter material deteriorates rapidly in the presence of ammonia.

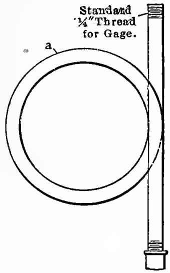


Fig. 11.—A Circular Siphon for Steam Gages.

Bourdon gages may be used for indicating the pressures of either liquids, steam or gases without observing special precautions if the temperature is never much over 150 degrees Fahrenheit. If, however, the elastic **tube** in the gage is heated above this limit it is likely to lose some of its temper. When used for steam pressure, therefore, **some form of siphon or water seal must always be used** to prevent steam from entering the gage. The type of siphon used most commonly is illustrated in **Fig. 11.** There is always a possibility, however, that air carried in the steam may be entrapped at **a**, where it forms a cushion, preventing the gage from indicating the true variation in pressure. For this reason the form of siphon shown in **Fig. 12** is preferred for accurate measurements.

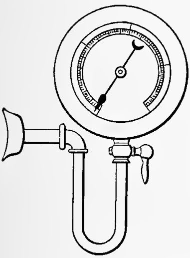


Fig. 12.—A U-shaped Siphon for Steam Gages.

In Bourdon gages any lost motion of the parts is taken up by the hair-spring attached to the spindle carrying the pointer.

Adjustments. The ratio of motion of the pointer with respect to that of the tube can be adjusted in most Bourdon gages by sliding a set-screw in a slot in the short arm of the rack-lever. In the gage illustrated in **Fig. 8** when the short arm of the rack-lever is made longer by adjusting the set-screw, the movement of the rack and also of the pointer is reduced for a given deflection of the tube.

Sometimes when used carelessly, especially when subjected to pressures beyond the scale on the dial, the tube of the gage takes a permanent " set "; or, in other words, it does not spring back to its original position, and the pointer does not come back to the zero mark. In such exigencies and also for adjustment after calibration the needle can be forced off from its spindle —preferably by the use of a clamp or " needle-jack " made by gage manufacturers specially for this service—and then set again in position where it should be.

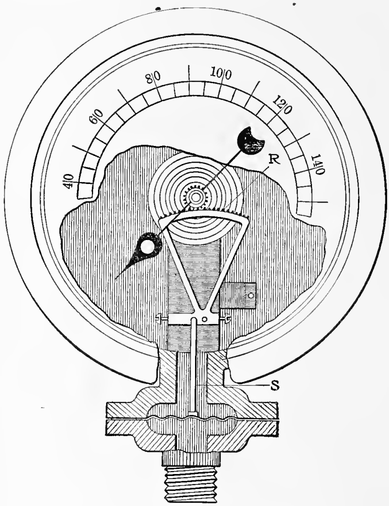


Fig. 13.—A Typical Diaphragm Gage.

Another kind of gage in which there is a metallic disk or diaphragm instead of a bent tube for actuating the indicating device is sometimes used. One of this type is well illustrated in **Fig. 13.** It consists of a **corrugated diaphragm** clamped around its edge by the flanges of an encircling chamber. Pressure applied on the lower side of the diaphragm deflects it upward, the amount of this upward movement being proportional to the pressure. By means of a connecting strut **S** the movement of the diaphragm is communicated to a rack **R** connected to a small pinion attached to the spindle of the needle indicating the pressure on the graduated dial.

Since the deflection of the center of the diaphragm is proportional to the pressure and is inversely proportional to the cube of its thickness, a very slight alteration in the thickness of the diaphragm will cause a considerable change in the reading of the gage.

Vacuum Gages. For the measurement of vacuum instead of pressure Bourdon gages are also very commonly used. The design used for a pressure gage is altered only in the arrangement of the levers moving the needle, which for vacuum measurements turn it in the same direction as for pressure (clock-

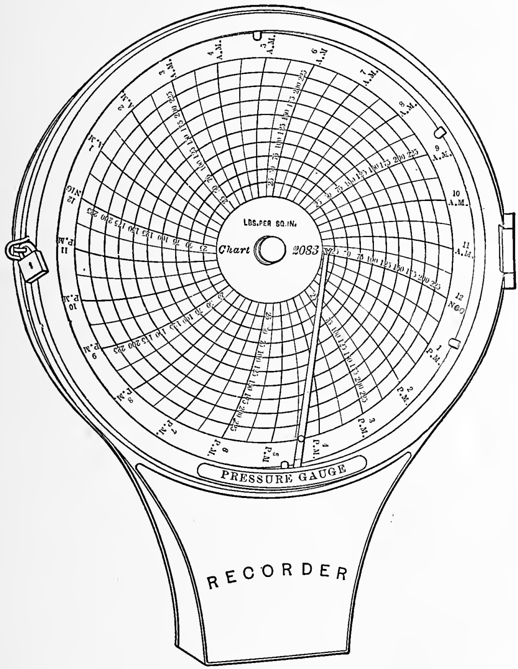


FIG. 14.—A Recording Pressure Gage.

wise), when, as in this case, the tube is bent inward or toward the center of the gage instead of outward as for pressure measurements. Vacuum gages are usually graduated to read **inches** of mercury below atmospheric pressure. **Absolute pressure in inches of mercury** is the difference between the barometer and the reading of such vacuum gages.

Still another type of pressure gages known as a **compound gage** is used to indicate either **pressure or vacuum on** the same dial.

Recording Gages. In many modern power plants recording gages are used to give a graphic record on a chart of the pres-

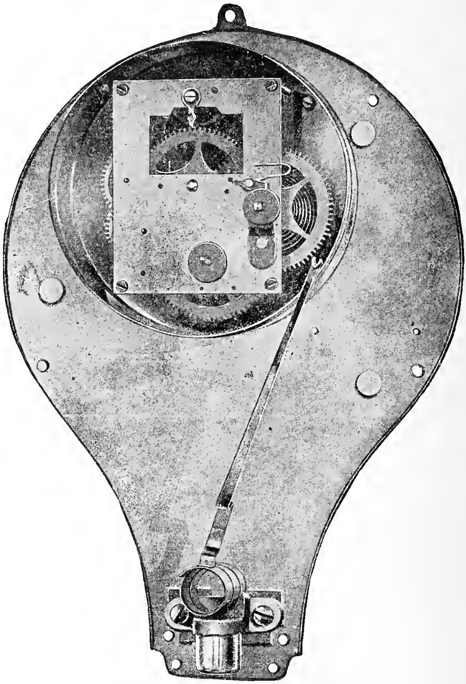


Fig. 15.—Operating Parts of a Recording Gage with a Helical Tube. (Bristol.)

sure or vacuum for 24 hours. The most common type of recording gage is shown in **Fig. 14.**

These gages are made with either a circular tube of oval section in the form of a helix as illustrated in **Fig. 15,** with a metallic Bourdon tube as shown in **Fig. 16,** or with a dia-

phragm device as in **Fig. 17.** The first and second of these three types are generally used for cases where the maximum pressure is greater than 3 pounds and the third when it is less. The recording arm is preferably attached directly to the moving element so that no gears, levers, or other multiplying devices are needed. A more compact and less expensive form of such gages is illustrated in **Fig. 18.**

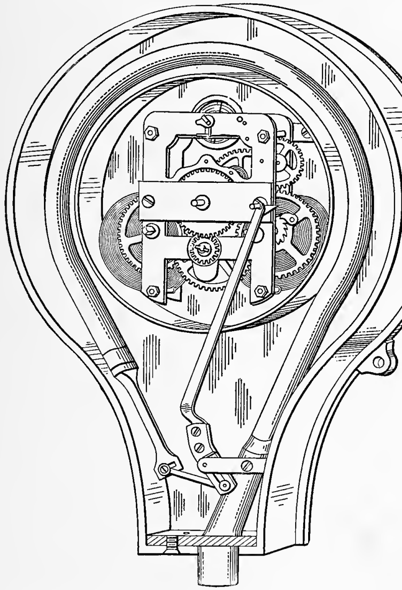


Fig. 16.—A Recording Pressure Gage Operated by a Bourdon Tube.

The **average pressure** corresponding to an irregular curve traced on the circular card of one of these recording gages is obtained with a fair degree of accuracy by integrating the curve by means of a Durand-Bristol integrating instrument described on page 80. Corrections to be applied to the readings of these gages are of course obtained by calibrating in the same way as for an indicating gage.

Still another type of recording pressure gages **is shown in Fig. 19.**

Calibration of Gages. Until recent years when the so-called "dead-weight" apparatus for testing gages came into general

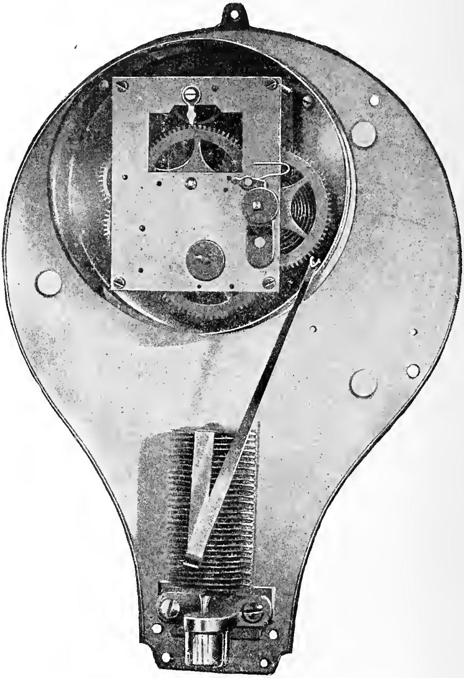


Fig. 17.—A Low-pressure Recording Gage Operated by a "Diaphragm" Device.

use, gages used in other places than engineering laboratories were commonly calibrated by **comparison** with a so-called **test gage.** Such test gages have usually somewhat finer graduations than the ordinary gages used in practice and are probably also adjusted a little more accurately. They should never be exposed to the severe conditions of service, being intended

only for purposes of comparison. This comparison can be made anywhere by connecting the standard and the gage to be tested to any system of piping in which the pressure can be varied either by pumping a liquid, or by means of valves "throttling" steam, water or air under pressure. The only important precaution to observe is that the two gages shall be at approximately the same level when a **liquid** is used, and that the velocity of the fluid in the main pipe to which the gages are attached is negligible or is the same at the points where the connections for the gages are inserted in the "main" pipe. Test gages must, of course, be calibrated from time to time with some standard apparatus to insure their accuracy. A bench pump suitable for calibrating by comparison is illustrated in **Fig. 20.**

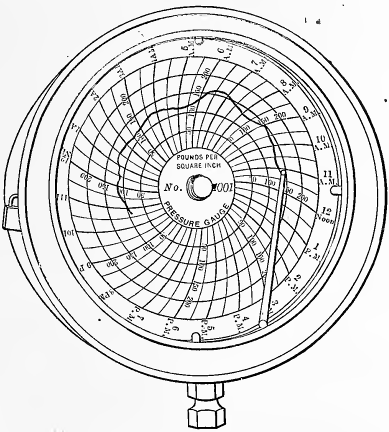


Fig. 18.—A Veyr Compact Tpye of Recording Gage. (Bristol)

Gage Testers. In very many power plants the use of the test gage has been superseded by some form of gage tester and by this means the gages used in the plant can be calibrated **directly** with an absolute standard. Calibrations of gages for high pressures by means of mercury columns are for practical reasons suitable only for laboratory work.

Dead-weight Gage Testers. The best-known form of this apparatus is made by the Crosby Steam Gage and Valve Co., and is illustrated in **Figs. 21** and **22.** The latter figure shows a partial section. It consists of a vertical cylinder **C,** into which is fitted very accurately a piston **P,** of which the area, **when new,** is exactly one-fifth of a square inch. A circular platform upon which weights can be placed is attached to the

upper end of this piston. The cylinder, **C,** communicates at its lower end with the reservoir **R,** fitted with an adjustable plunger working in a screw and is operated by a hand wheel. A pipe, **T,** attached to the lower part of the reservoir is provided with unions and special fittings for attaching gages of various sizes. In the horizontal portion of this pipe a three-way cock or valve **V** is provided for either draining the reservoir or for closing the pipe so that the liquid in the apparatus

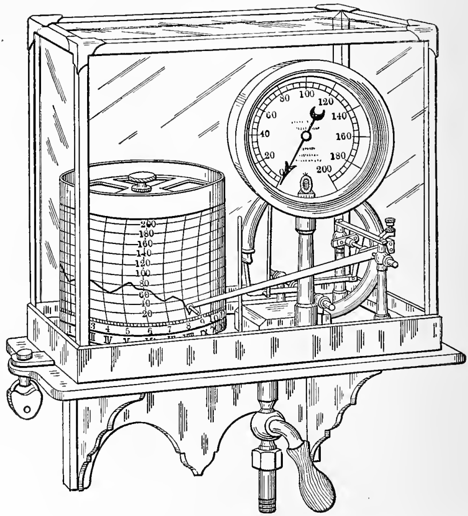


Fig. 19.—A Combined Recording and Indicating Pressure Gage.

will not escape when the gage is removed. In operation, after the gage has been attached securely, the adjustable plunger **S** (**Fig. 22**) is screwed down to the bottom of the reservoir **R,** then with the piston removed, glycerine or heavy oil is poured into the cylinder **C** at the same time that the plunger **P** is screwed out. In this way the reservoir can be completely filled with oil witout entrapping any considerable amount of air which would act as a cushion preventing the most satisfactory operation of the apparatus.

If the area of the piston is one-fifth of a square inch then each pound weight added on the platform produces a pressure on the liquid of 5 pounds per square inch. The weight of the platform and piston (usually 1 pound) must always be included in the weight producing the pressure. As the load on the platform is increased the plunger must, from time to time, be screwed in to keep the piston and platform "floating." When observations are being taken it is very essential that the loaded platform be given, preferably by hand, a slight rotary motion to reduce to a minimum the friction of the piston in its cylinder.

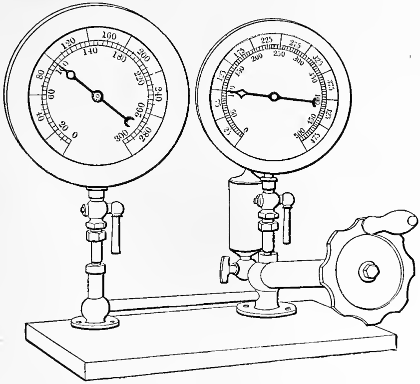


Fig. 20.—A Bench Test Pump.

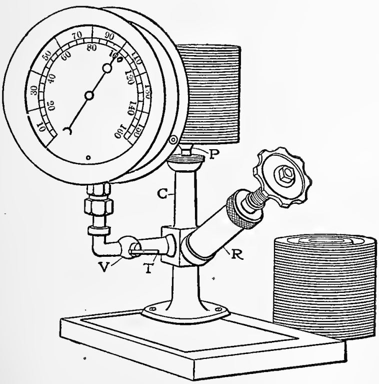


Fig. 21.—Crosby Dead-weight Gage Tester.

Suggested Procedure with Dead-weight Testers. The accuracy of the gage to be calibrated is determined by subjecting it to known pressures and noting its error. Before the piston has been put into place the reading of the gage, called "zero-reading," should be observed and recorded in a

form similar to the one on page 20. Then the pressure should be increased 5 pounds per square inch at a time (corresponding usually to a **weight** of 1 pound) up to the limit of the graduations on the dial, **spinning the piston gently when each reading is taken.** Commencing then with the highest pressure the

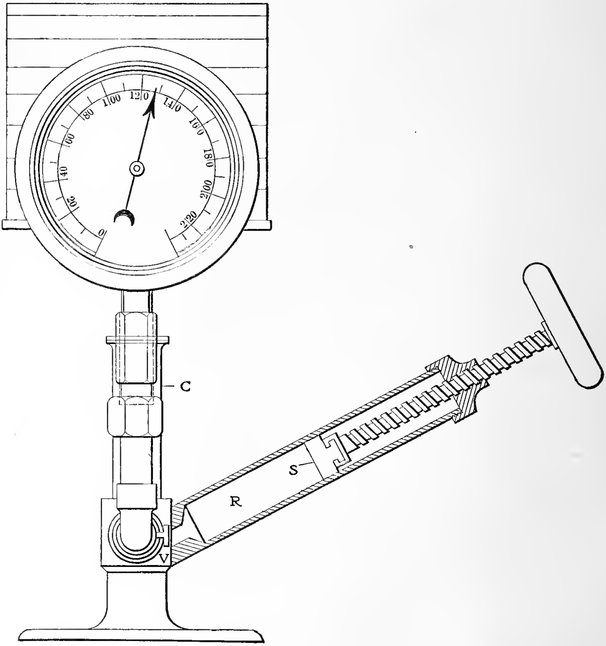


Fig 22.—Sectional View of a Dead-weight Gage Tester.

same operation should be repeated by decreasing the pressure by the same increments.[1]

[1] When the pressure is being decreased the movement of the pointer must be always in a counter-clockwise direction just before a reading is taken. In other words, if a weight of 2 pounds has been taken from the load on the piston when only 1 pound should have been removed the pointer will, of course, get below the next point to be calibrated. To secure the reading missed it will not be correct to add 1 pound and

A modification of the dead-weight gage tester is shown in **Fig. 23.** This instrument is particularly suited for calibrations at high pressures. Its range is from 0 to 1500 pounds per square inch.

Any pressure within these limits can be obtained without shifting heavy weights. Readings are taken when the scale beam is balanced. The hand wheel **A** is used to regulate the fluid pressure by means of a plunger as in the apparatus shown in **Figs. 21** and **22.** The other handwheel **B** shown in the figure must be kept rotating when observations are taken. The slight jarring of the parts due to its rotation serves to make the friction as small as possible. For still higher pressures up to 12,000 pounds per square inch, a heavy stationary type shown in **Fig. 24** can be used.

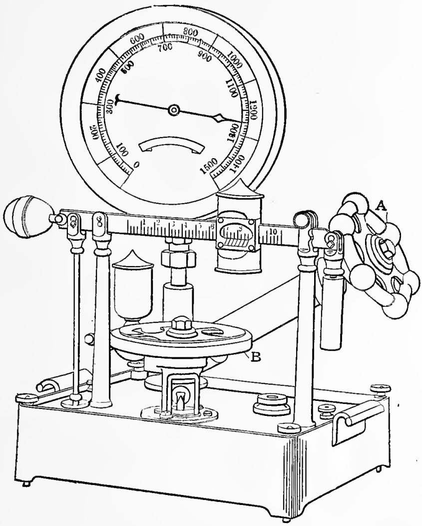


Fig. 23.—Crosby Portable Fluid Pressure Scales.

take the reading, because the friction and lost motion will now be in the same direction as with increasing pressures; and to overcome this difficulty the pressure must be increased again to a value higher than that for which the reading is to be taken. For the purpose of increasing the weight it is not necessary to put on more weights, as additional load in such cases can be put on by the pressure of the hand.

The same precautions apply with even greater force to calibrations made with test gages or with a mercury column. With either of these instruments discrepancies may occur with increasing or decreasing pressures. In fact the only certain way to get satisfactory results with these instruments is to keep the pointer of the gage or the mercury column, as the case may be, moving continually in the same direction.

Calibration of Gages with Mercury Columns. The ultimate standard for the determination of reasonably high pressures is the mercury column, but the apparatus required is so complicated and occupies so much space that this method is suitable only for use in laboratories where it will have the attention of skilled observers.

For the purpose of calibrating steam gages mercury columns have been fitted up in a variety of ways. Simplest of these is the method of connecting the gage to be tested by means of a short tube to a "**closed**" mercury well into the top of which a long glass tube has been inserted. The pressure can then be increased either by displacing some of the mercury in the well by means of the plunger in the mercury pump shown atth e right-hand side of **Fig. 25,** and forcing it up into the glass tube, or else by pouring mercury into the tube from the top as must be done in the apparatus shown in **Fig. 26.** Zero pressure for comparison is to be taken on the column at the same level as the center of the gage. Beginning then with 5 pounds per square inch pressure on the gage observe the corresponding height of the mercury column and its temperature, and then continue the observations, first increasing the pressure and then decreasing by increments of 5 pounds, as indicated by the gage. Equivalent units for calibration can be computed from the height of the mercury column, since 1 inch of mercury at 70 degrees Fahrenheit is equivalent to a pressure of .4906 pound per square inch.

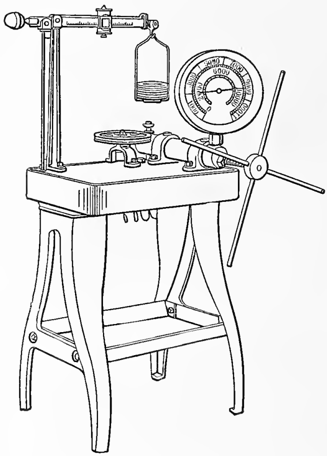


Fig. 24.—Fluid Pressure Scales for High Pressures.

When using a mercury-testing apparatus it is necessary to observe the temperature near the mercury column in the

room in which the work is being done, so that the observed height of the mercury column can be corrected to a temperature at which the relation between pressure in pounds per square inch and height is known. The coefficient of **cubical** expansion of mercury is not constant, as will be observed from the following table:

Temp. Deg. Fahr.	Coefficient of Cubical Expansion.
32	.0000998
50	.0001000
70	.0001002
90	.0001004
110	.0001007

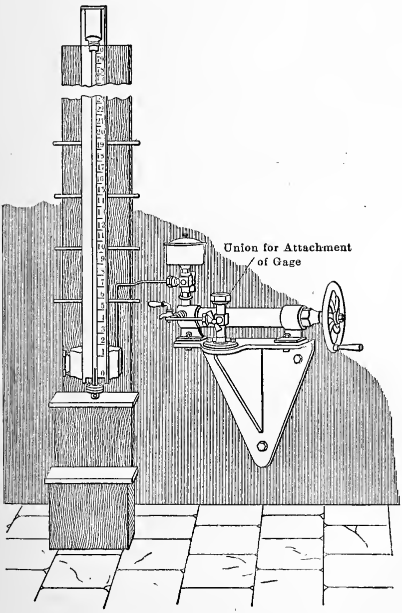


FIG. 25.—Standard Mercury Column and Hand Pump.

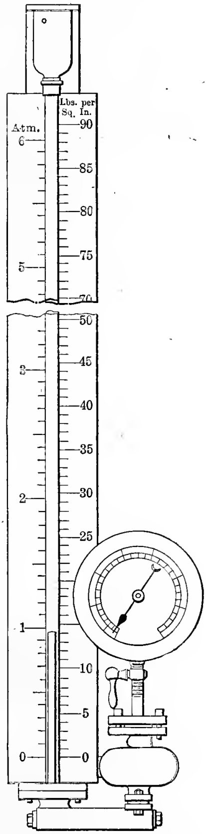


FIG. 26.—Simple Open Mercury Column.

For very accurate work allowance must be made for the expansion of the graduated scale. Coefficients of expansion of metals are given in the Appendix.

Instead of connecting the gage directly to the mercury well, it is sometimes connected to one end of a steam drum and the mercury column is connected to the other. The increments of pressure are then obtained by increasing or decreasing the steam pressure in the drum.

Observations taken in the calibration of a steam gage should be recorded and the computed errors tabulated in a form similar to the following:

CALIBRATION OF PRESSURE GAGE. COMPARISON WITH GAGE TESTER

Date.......................... Observers...................
Maker of gage................. Maker's No....................
Laboratory No.............. Limits of Graduation...............

No. of Reading.	Weight on Tester Lbs.	Gage Readings. Lbs. per sq.in.			Actual Pressure. Lbs. per sq.in.	Mean Error of gage. Lbs. per sq.in.	Remarks
		Up.	Down.	Mean.			

The error of the gage is determined by the comparison of the mean of the up and down readings with the **actual pressure.**

Curves. From the data tabulated two curves are usually plotted:

1. **Mean** gage readings (abscissas) and actual pressures (ordinates). Use a large sheet of coordinate paper for this curve.

2. **Error Curve: Mean** Gage Readings (abscissas) and mean corrections, positive and negative (ordinates). See curve in

Fig. 27. **Error curves** should have the points, if very irregular, connected by a broken line rather than by a " fair " or average curve through them. Never, however, try to draw an irregular curve through each of a number of scattered points when the points are supposed to follow a definite relation between the coordinates selected. A " fair " curve should then be drawn **between** the irregular points.

Calibration of Vacuum and Low-pressure Gages. A vacuum gage is usually calibrated by connecting it to one end of a U-shaped glass tube of which both legs are about 30 inches

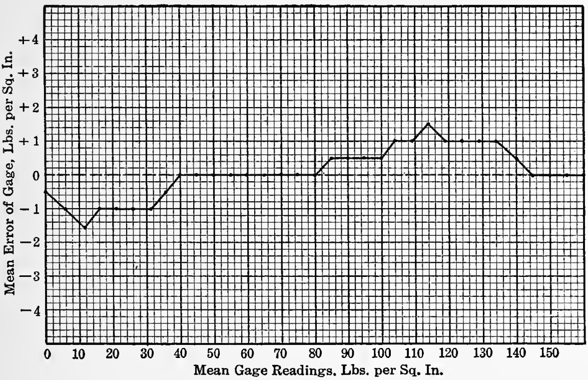


FIG. 27.—Typical Error Curve for a Pressure Gage.

long and are filled to about half their length with mercury. The U-tube and gage are then connected to the receiver of an air pump or else to an aspirator or ejector operated by water or steam pressure, such as chemists use for vacuum filtering. The aspirator is really the more convenient instrument to use. If the readings of the vacuum gage are correct, they will correspond exactly with the difference in the level of the mercury in the two legs of the U-tube.

In case a condensing engine is operating when the calibration of the vacuum gage is to be made, both the gage and the glass U-tube may be connected to the condenser. A comparison of the readings taken will show, under the best possible

conditions, the absolute errors of the gage. A suitable scale about 30 inches long and accurately graduated should of course be provided and placed between the two legs of the U-tube.

An apparatus consisting of an air-pump designed for a high vacuum and a mercury column is illustrated in **Fig. 28.** It is a very convenient means for testing vacuum gages.

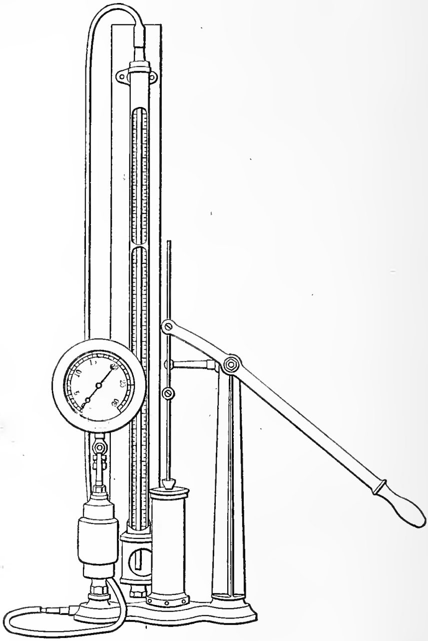


Fig. 28.—Air Pump and Mercury Columns for Testing Vacuum Gages.

A low-pressure gage with a scale from say 0 to 15 pounds per square inch is very easily and accurately calibrated by using the same glass U-tube mentioned for the calibration of the vacuum gage with air **pressure,** preferably, or with steam pressure. Otherwise the method for calibration is the same as for a vacuum gage, except that inches of pressure instead of inches of vacuum are observed.

Draft Gages. Many engineers use an ordinary glass U-tube manometer (**Fig. 29**) filled with water for measuring small pressures like that due to the draft in a chimney or that produced in air-ducts by ventilating fans or blowers. For such observations in many cases, however, greater accuracy is desired than can be secured by the use of the ordinary U-tube and a special form of manometer is used in which the distance moved by the surface of the liquid in the tube is greater than the vertical change of level. **Fig. 30** illustrates a simple device of this kind. It consists of a bottle **B,** filled with water, having a suitable opening at the bottom to which by means of a short rubber tube the inclined glass tube **CD** is attached. At the upper end of this tube a piece of rubber tubing **T** is shown and is intended to be connected to the chimney, duct, or flue in which the pressure is to be obtained. A scale placed behind the inclined tube **CD** should be graduated so that when the spirit level **L** is adjusted, **vertical** differences in level in the bottle will be indicated by the scale. Then differences in the readings of the scale will give directly the difference in pressure in inches of water just as with an ordinary U-tube.

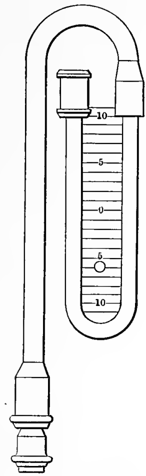


FIG. 29.—Simple U-tube Draft Gage.

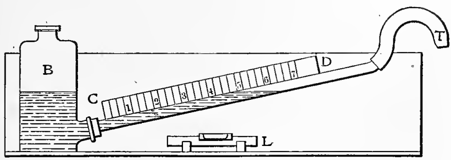


FIG. 30.—Inclined Tube Draft Gage.

Very accurate draft gages of this type known as Ellison's, are shown in **Figs. 31** and **32.** The inclination of the tube in these instruments is usually about 1 to 10. Instead of water a very light oil is used to fill the tube. It is claimed that this

oil has the advantages of having less capillarity than water and also, being lighter, permits the use of a longer scale for a given difference in level. Graduations on these instruments which are sold commercially[1] are, however, always made to

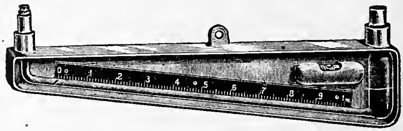


FIG. 31.—Ellison's Improved Draft Gage.

read equivalent inches of **water.** These draft gages are also often used for measuring small differences of pressure. For example, if there are two vessels containing gases at different pressures and one is connected to the left-hand side and the other to the

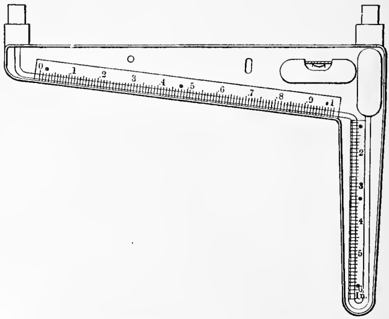


FIG. 32.—Ellison's Differential-direct Draft Gage for High Drafts.

right-hand side of the gage, it will indicate the difference in pressure.

When calibrating gages it is worth while to notice that when instruments are to be used to observe practically constant values it is necessary to calibrate them only near the values to be observed.

[1] American Steam Gage and Valve Mfg. Co., Boston and Chicago.

CHAPTER II

MEASUREMENT OF TEMPERATURE

Mercurial Thermometers. Temperatures less than about 500 degrees Fahrenheit are usually measured by means of mercurial thermometers, depending for their action on the expansion of mercury in a glass bulb and a graduated capillary tube.[1]

Whenever mercurial thermometers are used for any work where reasonable accuracy is expected they should be carefully calibrated before the test is made; and after the test the calibration should be at least roughly checked to be sure that the zero of the thermometer has not changed. Too often it happens in practice when tests are being made, as for example of a boiler or of a steam turbine, that in some way a thermometer not previously calibrated has been used, and before the end of the test is broken. It is then too late to get a calibration and sometimes very important results of tests are made doubtful because of such negligence.

Calibrations of thermometers of all kinds must be made often because there is always the possibility that a little of the mercury has become detached from the column and remains unobserved either on the sides or at the top of the capillary tube. In all glass thermometers there is always taking place with use and time a gradual and permanent change in the

[1] The best thermometers for ordinary engineering work are those having graduations etched on the tube. The only difficulty with this method is that after considerable use the ink originally in the etched markings disappears and it becomes difficult to read the scale. When this happens a thick paint made of lampblack and shellac or printer's ink can be rubbed over the etched scale, and when this paint is rubbed off after a few minutes, there will be enough of it left in the etchings to make the scale as legible as when new. A crayon or pencil of soft greasy graphite like those used by glaziers for marking glass or by shippers for marking cases is a satisfactory substitute for the paint, although it is not so permanent.

volume of the bulb, more in new thermometers than in old ones, altering the zero point, and of course, also the true values for all the graduations.

Alcohol Thermometers. For the measurement of temperatures much below zero Fahrenheit thermometers filled with mercury are not satisfactory, and alcohol or " spirits of wine " is used. These liquids, on the other hand, are not suited on account of their high vapor tensions for high temperatures.

Conversion of Temperatures and Heat Units. Temperatures in Centegrade degrees are converted into Fahrenheit by multiplying by $\frac{9}{5}$ and adding 32. Kilogram-calories multiplied by 3.968 give the equivalent British thermal units (B.T.U.), and kilogram-calories per kilogram $\times 1.8$ give British thermal units per pound. A " small " or gram calorie is one-thousandth as large as a kilogram-calorie.

Calibration of Thermometers. Tests to determine the accuracy of thermometers are made by subjecting them to known temperatures and noting the errors. This is done usually in one of two ways:

1. By comparison with a so-called " standard " thermometer known to be accurate.

2. By comparison with temperatures corresponding to steam pressures.

Since the second method is not applicable for temperatures below the boiling point of water, it is not often used for temperatures below 212 degrees Fahrenheit. For " low-reading " thermometers, therefore, the first method is generally used.

Calibration by Comparison with a Standard Thermometer. For low-temperature calibrations, the thermometers to be tested are usually suspended together with a " standard " thermometer of which the errors are known in a water bath arranged so that the temperature can be varied. This bath may consist simply of a vessel provided with a coil of pipe through which steam can be circulated, and has also a suitable stirring device. If the water is kept well stirred in such an apparatus a uniform temperature can be maintained and three or four thermometers can be calibrated at the same time.

Fig. 33 illustrates diagrammatically a very simple apparatus of this kind, except that the water bath is heated by discharg-

ing steam directly into the water. This arrangement permits changing the temperature more rapidly than with the coil of pipe mentioned above.

"Standard" thermometers for comparison should be preferably those which have been calibrated at standardizing laboratories such as at the U. S. Bureau of Standards at Washington, D. C., at the Reichsanstat at Berlin, Germany, or at the Royal Physical Testing Laboratories in London, England.

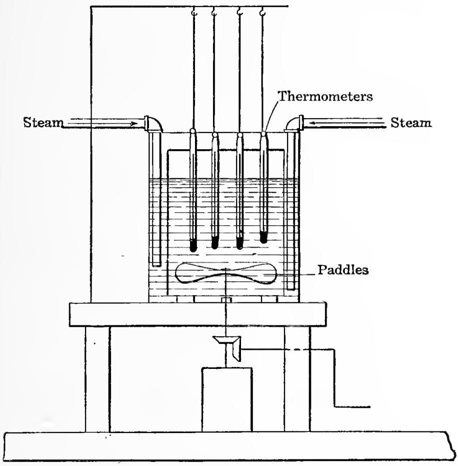


FIG. 33.—Apparatus for Calibration of Thermometers at Temperatures Less than the Boiling-point of Water.

The steam laboratories of nearly all technical colleges have sets of standard thermometers suitable for determining the errors of good thermometers to be used as "secondary" standards.

When the method of comparison with a "standard" thermometer is to be used for temperatures higher than are obtainable with the apparatus shown in **Fig. 33**, the **"standard"** thermometer and the other thermometers to be calibrated are placed in adjacent cups or wells of the same depth inserted in a suitable cylindrical drum with pipe connections permitting

a flow of steam through it. The thermometer cups or wells should be filled with cylinder oil or, preferably, for high temperatures, with mercury.[1] Temperature is varied by throttling with the valves on either or both the steam inlet and discharge pipes. Usually the necessary adjustment is made more easily by manipulating the discharge valve rather than the inlet. At least **five minutes** should be allowed after the valves have been adjusted for the mercury in the thermometers to come to rest before readings for comparison are taken. Readings of both the standard and the thermometer being calibrated must be taken as nearly as possible at the same time, and the thermometers should be lifted from their cups when necessary just enough to bring the mercury into view. Observations should be taken as quickly as possible to avoid errors due to cooling and should be made with approximately the same increments.[1]

A record of the observations should be made in the form given below.

CALIBRATION OF THERMOMETER

By comparison with " STANDARD "..............................
Date.................... Observers ....

Standard: **Thermometer Tested:**

No. or distinguishing mark.........No. or distinguishing mark..........
Range of scale...................Range of scale..................

No.	Standard Thermometer.		True Temperature.	Thermometer Tested.		Remarks.
	Reading. ° F.	Known Error. ° F.	° F.	Reading. ° F.	Error. (+ or −) ° F.	

[1] If oil is used in thermometer cups, precautions should be taken that the oil is absolutely free from the presence of water and that there is no water in the cups. If water is allowed to accumulate there

Calibration of Thermometers by Comparison with Temperatures Corresponding to Steam Pressures. There is a definite temperature of saturated steam corresponding to every pressure. If, then, the pressure is known, the temperature corresponding can be obtained from **"Tables of the Properties of Saturated Steam."**[1]

Thermometers to be calibrated are placed in adjacent cups or wells of the same depth inserted in a cylindrical steam drum and filled with cylinder oil or mercury. Pipe connections must be provided for the attachment of an accurately calibrated steam gage and valves are needed at the ends of the drum for regulating the flow of steam through it. Except for the addition of the gage the apparatus is the same as that used for the calibration of thermometers at high temperatures explained in the preceding paragraphs, and the manipulation of the apparatus as well as the precautions to be observed are also the same except that the pressure registered by the gage is recorded instead of the temperature indicated by the "standard" thermometer.

If there is any possibility that the steam supplied to the drum is superheated, then it is necessary to provide a water-jacket around the steam pipe large enough to make the steam at least dry saturated or preferably slightly wet. Another device often used to change superheated steam to the saturated condition is illustrated in **Fig. 34.** In this apparatus the steam passes down through the vertical supply pipe **S** closed at the lower end and escapes from perforations near the bottom to bubble up through the water contained in the chamber **A** and is carried away in the pipe **D.** In this way the steam can be made to lose enough heat to the water to reduce the superheat. A gage glass **G** at the side of the drum is serviceable for showing the level of the water.

may be a slight explosion, sometimes strong enough, however, to throw the thermometer out of the cup.

Thermometers used for measuring high temperatures should be protected from contact with the metal of the cups by wrapping the stems at the mouth of the cup with cotton waste, or, preferably, with rings of cork.

[1] Marks and Davis' *Steam Tables and Diagrams* or Peabody's *Steam and Eutropy* Tables (revised 1909) are recommended.

A most excellent method for making calibrations of thermometers by means of a steam drum is a combination of the last two methods described. That is, the corrections are calculated from the temperature corresponding to the corrected gage pressure; but at the same time are **checked** by comparison with a "standard" thermometer.

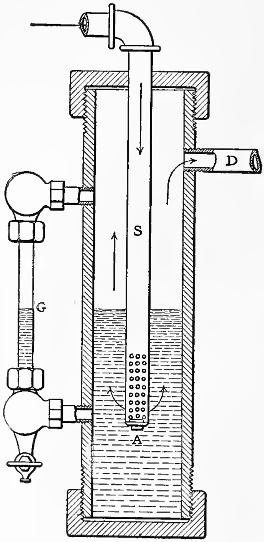


FIG. 34.—Apparatus to Reduce the Superheat in Steam.

In all experimental work in engineering such methods of checking results cannot be too highly commended. Checking not only assists in the elimination of errors of observation as well as in calculations, but for the engineer it is his key to success.[1]

Observations should be tabulated in the form given on page 31. The column under the heading "Standard Thermometer" is to be left blank, of course, when only the steam gage is used for the calibration. To obtain the absolute pressure the barometric pressure must be observed during the calibration.

Curves. 1. Plot a curve for each thermometer showing observed temperatures of the thermometer tested (abscissas) and the corresponding "true" or "standard" temperatures (ordinates). This curve is of no value, however, unless a reasonably large scale is used.

2. Plot an **error curve,** taking observed temperatures of thermometer tested for abscissas and "errors" for ordinates. Compare with **Fig. 27.**

[1] The **plotting of curves** is still another means which the engineer uses continually for checking his calibrations, his tests, and his conclusions. It has been well said that "the physician buries his mistakes but that the mistakes of the engineer bury him."

CALIBRATION OF THERMOMETER BY COMPARISON WITH TEMPERATURES CORRESPONDING TO STEAM PRESSURES

Record:

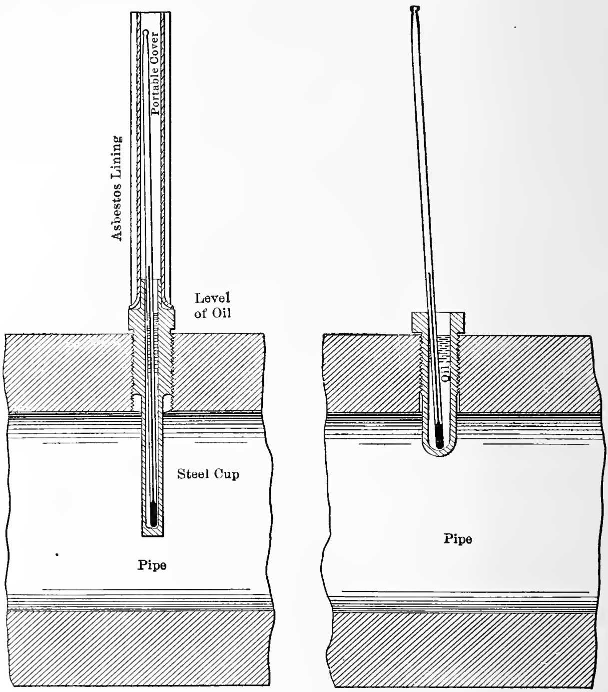
1. Date and names of observers.................................
2. No. and type of gage...
3. No. of standard thermometer..................................
4. Identification of thermometer tested.........................
5. Limits of graduation of both..................................
6. Barometer reading..............ins. mercury.................

1	2	3	4	5	6	7	8	9	10	
		Observed Temperatures ° F.		Pressures Lbs. per Sq.In.						
No. of Reading.	Time of Reading.	Standard Thermometer.	Thermometer Tested.	Gage Readings.	Barometric.	Corrected Gage Readings.	Absolute (6)+(7).	True Temp. from (8) and Steam Table ° F.	Error of Thermometer Tested Compared with (9) °F.	Remarks.

Experience has shown that certain types of thermometer cups or wells for use in pipes give more satisfactory results than others. The thermometer cup must be long enough to enter well into the pipe so that the flow of fluid through it will be around the well. In other words it should be located so that it will be in the "main stream" and not in such a position where only eddies touch it. A well-designed thermometer cup is illustrated in **Fig. 35**.

Correction for "Stem Exposure" of Mercury Thermometers. When a considerable portion of the mercury column of a thermometer measuring high temperatures is exposed to the air, a correction **K** must be added to the readings to ob-

tain the true temperature. If **t** is the observed reading, **D** is the number of degrees Fahrenheit on the scale from the surface of the oil or mercury in the thermometer cup to the



The Right Way. The Wrong Way.

FIG. 35.—Typical Thermometer Wells—" Good and Bad."

end of the mercury column, and **t′** is the temperature of the air surrounding the thermometer stem, then

$$K = .000,088D(t - t'). \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

This equation, it will be observed, includes three factors: coefficient of expansion, length, and temperature difference. The

coefficient of expansion given in the equation is the difference between the **volumetric** coefficient of expansion of mercury (.000,100) [1] and the linear coefficient (.000,012) of the kind of glass ordinarily used for thermometers.

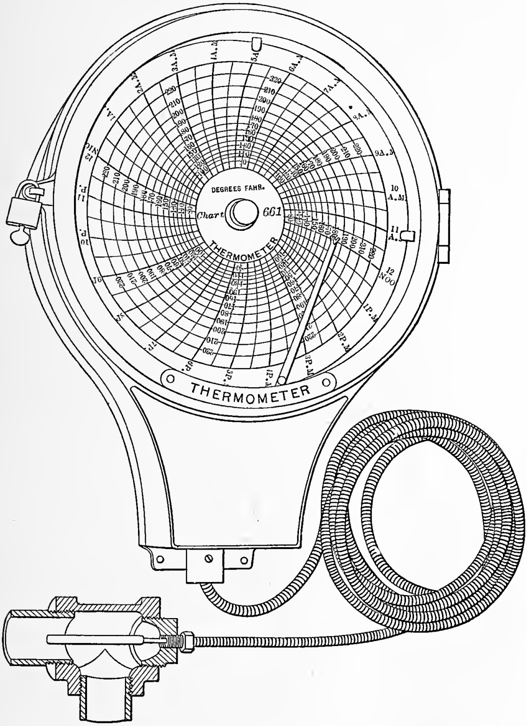


Fig. 36.—Typical Recording Thomometer with Flexible Tube.

In practice for carefully conducted tests of engines or turbines operating with superheated steam the above correction

[1] An average value of the volumetric coefficient is used here. A table showing the variation of this coefficient is given on page 19.

should always be added to the thermometer readings to obtain

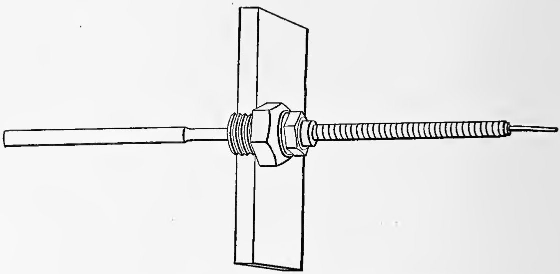


FIG. 37.—" Sensitive " Bulb for a Recording Thermometer.

the correct temperature and superheat. In steam turbine tests, when a high degree of superheat is used, this correction is often as much as from 5 to 10 degrees Fahrenheit.

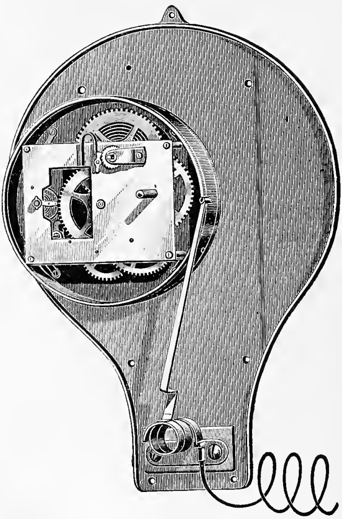


FIG. 38.—Mechanism of a Recording Thermometer.

Recording Thermometers. Recently instruments for recording automatically low as well as high temperatures have been very satisfactorily developed. A typical example is shown in **Fig. 36.** It consists of a sensitive bulb (**Fig. 37**) suitable for being inserted into a pipe fitting and is attached by a capillary connecting tube to the recording instrument. The sensitive bulb and capillary tube are filled with either alcohol or ether, which is sealed in the bulb and tube under pressure. The instrument is operated by the expansion of the vapor of these

liquids. One of these instruments is shown in **Fig. 38** with the cover removed so that the mechanism can be seen. It is exactly the same as that of a recording pressure gage (see page 10).

In general appearance these recording thermometers are like the recording pressure gages now in general use, and in

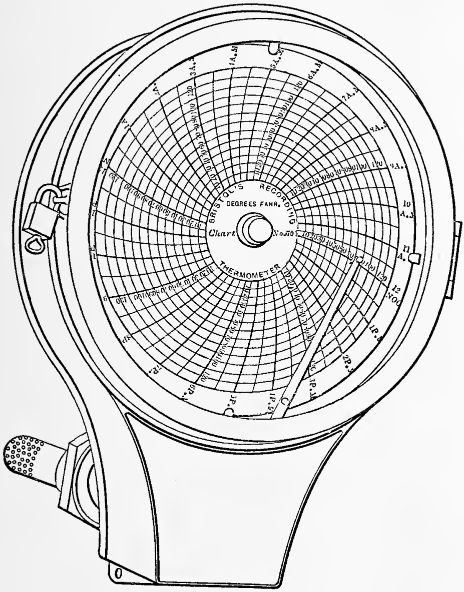


FIG. 39.—Recording Thermometer, Short Bulb Type.

other respects as regards the recording mechanism and clock-work they are also practically the same. Movement of the recording pencil proportional to the temperature is secured in most instruments of this kind by the expansion of ether or mercury in the "sensitive" bulb inserted usually by screw threads in the pipe or chimney where the temperature is to be obtained. Some of these recording instruments, **Fig. 39,** have a short rigid

connection between the bulb and the recording mechanism, making it necessary to locate the instrument always immediately adjacent to the bulb. In **Fig. 36** there is a flexible connection of capillary tubing attached to bulb permitting the setting up of the instrument on a wall near by. This capillary tube must, however, be handled very carefully to prevent causing a serious leak, making the instrument useless.

Pyrometers. Temperatures over 600 degrees Fahrenheit are usually measured by instruments known as pyrometers. Various types are in use particularly for the measurement of temperatures in flues and chimneys of boiler plants. The **thermo-electric pyrometer** is probably the one most commonly used in modern plants and is doubtless the most reliable. One of the best-known makes is illustrated in **Fig. 40.** It consists of a twisted joint formed of wires of two different metals having such thermo-electric properties that when joined they generate an electric current capable of deflecting a sensitive galvanometer. When, therefore, this joint or "couple" is heated the electric current generated produces a deflection of the needle of the galvanometer in proportion to the temperature. The scale of the instrument is generally graduated to read directly in degrees of temperature. For instruments reading up to about 1500 degrees Fahrenheit the "couple" consists of a wire of nickel for the positive element and an alloy of nickel and chromium for the negative element, while if the thermo-electric joint is made of one wire of pure platinum and another of an alloy of platinum and about ten per cent of rhodium or of iridium, temperatures nearly as high as the melting point of platinum, or nearly 3500 degrees Fahrenheit, can be measured, although 3000 degrees Fahrenheit is considered the

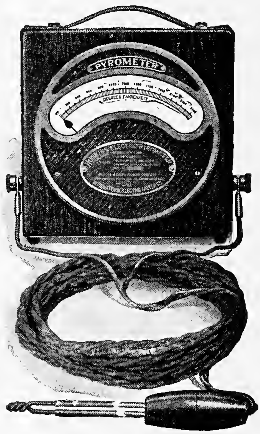


FIG. 40.—Thermo-Electric Pyrometer.

safe limit. This pyrometer with a platinum "couple" is known as the Le Chatelier (**Fig. 41**).

Electric Resistance Pyrometers like the one illustrated in **Fig. 42** are sometimes preferred when a very sensitive instrument is desired. The "heating" element consists of a coil of very fine platinum wire wound on a mica frame. The current from one electric battery passes through this wire, and the current from another battery passes through the coil of "resistance" wire in the cover of the box. When the two circuits are connected so that the electromotive forces of the two batteries are opposed, the resistance in the cover is adjusted by means of a connection on a stylus so that there is no current passing through a telephone or a galvanometer placed at the junction of the two circuits. For making observations this stylus is moved along the "scale" wire in the cover to a point where the humming noise due to the electric current ceases. The temperature can then be read on the scale opposite the position of the stylus.

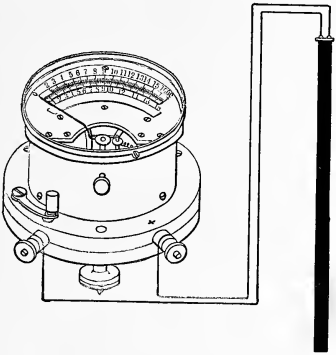


FIG. 41.—Le Chatelier Pyrometer.

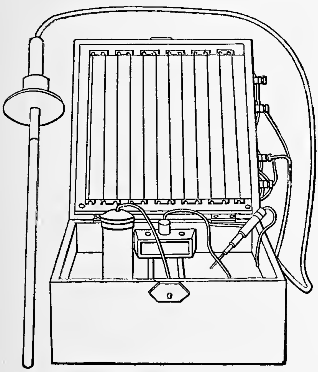


FIG. 42.—Electric Resistance Pyrometer.

By means of a switchboard any number of "heating" elements can be connected to the same indicator box, which may be located at any distance from the source of heat.

Mechanical Pyrometers consist of two metals having different rates of expansion, such as copper and iron or graphite and iron. By means of levers and gears the expansion is made to rotate a needle over a dial graduated in degrees. One of these instruments is illustrated in **Fig. 43.** It can be used safely to 1500 degrees Fahrenheit.

Calibrations of "Indicating" Pyrometers such as the **thermo-electric** and **metallic** are best made by comparison with a special standard pyrometer of which the error is known and which is used only for standardizing work. The couple to be calibrated and the standard should be fastened together closely with only a sheet of asbestos between them. The two couples thus bound together should be put into an electric furnace in which the temperature can be controlled and raised very slowly. Then at different points in the scale, at intervals of about fifteen minutes, readings for comparison can be taken. If a standard pyrometer is not available a calibration can be made by comparison in a furnace of constant temperature with a good mercury thermometer. Such thermometers in which the capillary tube contains rarefied nitrogen above the mercury can be obtained to measure temperatures with a fair degree of accuracy, when new, up to 1000 degrees Fahrenheit.

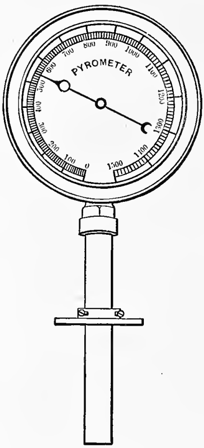


Fig. 43.—"Mechanical" Pyrometer.

Recording Pyrometers are most frequently of the type of recording thermometers illustrated and described on pages 33-35. Such instruments can be constructed, when the sensitive bulb is filled with a gas instead of a liquid, to register accurately temperatures as high as 1200 degrees Fahrenheit.

Another type operated by the expansion of the vapor of mercury is shown in **Fig. 44.** This is a combined indicating and recording instrument. The sealed tube **A** is to be inserted in the chimney or flue in which the temperature is to be observed.

Optical Pyrometers. For temperatures above 2500 degrees Fahrenheit optical pyrometers similar to the one illustrated in **Figs. 45, 46** and **47** are most suitable. They can also be used in many places where is it almost impossible to locate a pyrometer of any of the other types. It consists of a cylindrical case set upon a tripod. This case contains a concave mirror and a lens (or lenses) which when properly adjusted and focused on a hot body concentrate the heat rays upon a small thermo-electric couple inside the case. Copper wires connect this couple

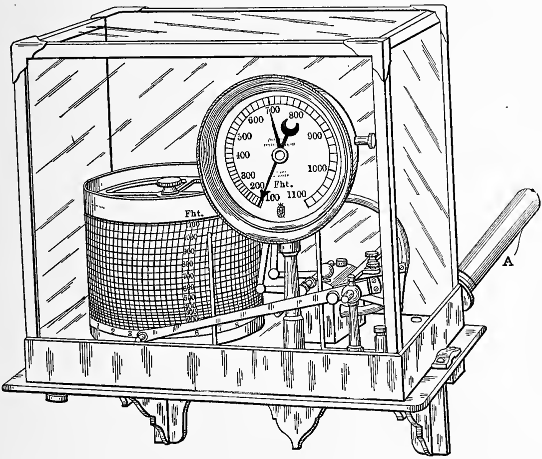


Fig. 44.—Combined Indicating and Recording Pyrometer.

with a very sensitive portable galvanometer (**Fig. 46**) located where it can be read conveniently. The most modern instruments of this kind are provided with scales indicating directly degrees of temperature. **Fig. 47** shows a section of the telescope used in connection with this pyrometer. The concave mirror **M** receives the heat rays and focuses them at **F,** where a small thermo-couple is located. To assist in pointing the telescope an eye-piece **E** is provided through which a reflected image of the hot body can be seen. The rack **R** and the pinion **P,** moved by a thumbscrew outside the case, serve

for adjusting the focus of the mirror. In the center of the field of view, as seen in the eye-piece, the thermo-couple is seen as a black spot, and this must be overlapped on all

FIG. 45.—An Optical (Radiation) Pyrometer in Use.

sides by the image of the hot body to obtain the correct temperature. It is interesting to observe that the distance of the telescope from the source of heat does not affect the reading of the instrument. When the telescope gets nearer the hot body the mirror **M** receives of course more heat, but at the same

time this greater amount of heat is distributed over a larger image and the intensity of the heat remains the same.

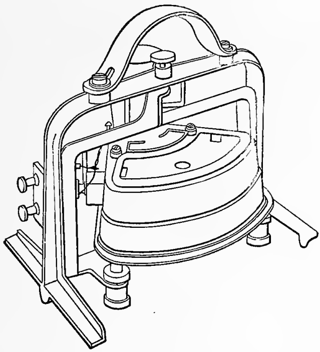


FIG. 46.—Sensitive Galvanometer of Fery Radiation Pyrometer.

Optical pyrometers are invaluable for determining the temperatures of the various parts of a furnace, of the walls of the setting of a steam boiler, of various portions of a bed of coals, etc.

Another type of optical pyrometer, based in principle upon the measurement of the brightness of the hot body by comparison with a standard lamp is shown in **Fig. 48.** In order to use this instrument, known as Wanner's, the incandescent (osmium filament) lamp must first be standardized by comparison with an amylacetate oil lamp of constant candle power. Then after standardizing it is only necessary to focus the instrument upon the hot body to be measured and the temperature is read directly on the graduated scale at the eye-piece.

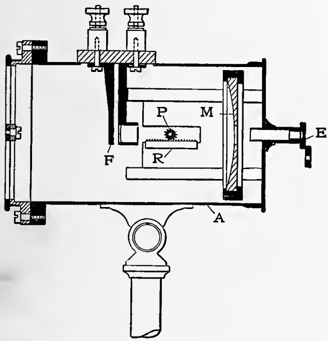


FIG. 47.—Telescope of Fery Radiation Pyrometer.

Calorimetric Pyrometers. If the specific heat and weight of a body are known, its temperature can be obtained by observing the rise in temperature of a known quantity of water into which the body is thrown.

More in detail the method consists in the determination of temperature by putting a ball of metal or other refractory

material into the medium of which the temperature is to be measured. When the ball has become heated uniformly throughout its mass to the temperature of the medium it is transferred quickly to a cup heavily jacketed with non-conducting material in which there is a known weight of water at a known temperature. Copper, wrought-iron and fire-clay are suitable materials. Specific heats of these materials at about 500 degrees Fahrenheit are respectively .097, .110 and .180. Since metals are readily attacked by furnace gases they should be protected when used in this way in a crucible of refractory material.

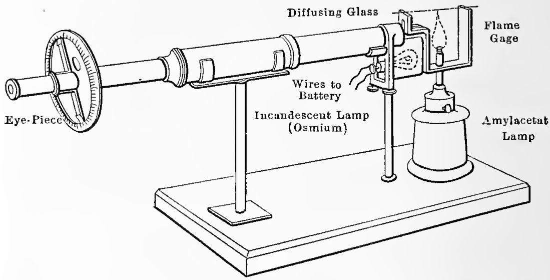


FIG. 48.—Wanner Optical Pyrometer in Position for Standardizing.

This method is often very serviceable in places or at times when accurate pyrometers are not available. On account of the "personal" error liable to enter, such determinations should be repeated several times to check the results. Calculations required are as follows.[1]

Let w_1 = weight of the ball, pounds.

w_2 = weight of the cup (only the "inner" vessel),[2] pounds.

w_3 = weight of the water in the cup, pounds.

t_1 = initial temperature of water, degrees Fahr.

t_2 = final temperature of the water, " "

[1] A more complete description of calorimetric pyrometers and the precautions to be observed for accuracy will be found in *Proceedings American Society of Mechanical Engineers*, vol. VI, page 712.

[2] It would be more accurate, of course, to use in the calculation the water equivalent of the whole vessel, as is done in coal calorimetry. See page 162. Units given are in pounds and degrees Fahrenheit, but other units, provided they are corresponding, can be used in the equation given.

Let t_0 = initial temperature of the ball, degrees Fahr.
s_1 = specific heat of the ball.
s_2 = specific heat of the cup.

Then
$$w_1s_1(t_0-t_2)=(w_2s_2+w_3)(t_2-t_1),$$
$$t_0=\frac{(w_2s_2+w_3)(t_2-t_1)}{w_1s_1}+t_2. \quad . \quad . \quad . \quad . \quad . \quad . \qquad (2)$$

Seger Pyrometer Cones. For many purposes when a pyrometer cannot be well placed fusible Seger cones are used. Such cones are made of several different oxides mixed in a manner to give a definitely known melting point for each one. The melting points range from 590 degrees to 1850 degrees Centigrade by steps of from 20 to 30 degrees, each having a standard number. These cones are carefully graded, so that if one

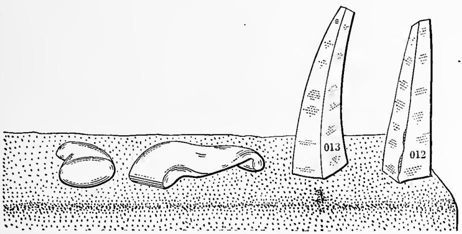


FIG. 49.—Seger Cones after Use.

has had some experience with them, temperatures can be **estimated** to about the nearest ten degrees in Centigrade. Four of these cones are shown in **Fig. 49.**

When a series of cones is placed in a furnace the one having the lowest melting point begins to turn over first. The temperature corresponding to the cone number is reached when the tip of the cone has bent over and just touches the surface on which it is standing. Hence the highest temperature reached when the cones shown in the illustration were used was about half way between that corresponding to each of the two middle cones. According to the numbers on the cones the temperature, as given by the table on page 45, was between 830 and 860 degrees Centigrade. The greatest disadvantage with this system is that there is no way of observing a decrease in the temperature, or, in other words, only the maximum temperature is recorded.

Two types of mercury thermometers protected by heavy metal cases are illustrated by **Figs. 50** and **51.** It will be observed that a very satisfactory thermometer cup is a part of the casing. The one shown in **Fig. 51** has graduations for reading both temperatures and pressures. A thermometer of this type is particularly useful in pipes carrying hot boiler feed-water. When the temperature is above 212 degrees

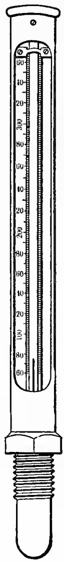


Fig. 50.—Combined Thermometer Cap and Protective Casing.

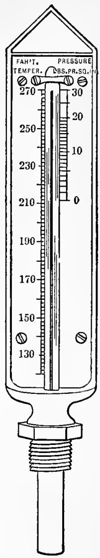


Fig. 51.—Combined Thermometer and Pressure Gage for Boiler Feed-water Pipes.

Fahrenheit the thermometer will indicate that the water is being heated at a pressure higher than atmospheric. For water heated in closed vessels or pipes there is for every temperature a corresponding pressure as given in tables of the properties of saturated steam.[1]

[1] Short and very much abbreviated tables of the properties of saturated steam are given in the Appendix. See also references in footnote on page 29.

The following table gives the temperatures, in degrees **Centigrade,** at which the Seger cones will begin to melt:

Seger Cone No.	Temp. Deg. C.	Seger Cone No.	Temp. Deg. C.	Seger Cone No.	Temp. Deg. C.
022	590	04	1070	15	1430
021	620	03	1090	16	1450
020	650	02	1110	17	1470
019	680	01	1130	18	1490
018	710	1	1150	19	1510
017	740	2	1170	20	1530
016	770	3	1190	......	....
015	800	4	1210	26	1650
014	830	5	1230	27	1670
013	860	6	1250	28	1690
012	890	7	1270	29	1710
011	920	8	1290	30	1730
010	950	9	1310	31	1750
09	970	10	1330	32	1770
08	990	11	1350	33	1790
07	1010	12	1370	34	1810
06	1030	13	1390	35	1830
05	1050	14	1410	36	1850

CHAPTER III

DETERMINATION OF THE MOISTURE IN STEAM

Unless the steam used in the power plant is superheated it is said to be either dry or wet, depending on whether or not it contains water in suspension. The general types of **steam calorimeters,** used to determine the amount of moisture in the steam, may be classified under three heads:

1. Throttling or superheating calorimeters.
2. Separating calorimeters.
3. Condensing calorimeters.

Throttling or Superheating Calorimeters. The type of steam calorimeter used most in engineering practice operates by passing a sample of the steam through a very small orifice, in which it is superheated by throttling. A very satisfactory calorimeter of this kind can be made of pipe fittings as illustrated in **Fig. 52.** It consists of an orifice **O,** discharging into a chamber **C,** into which a thermometer, **T,** is inserted, and a mercury manometer is usually attached to the cock $\mathbf{V_3}$, for observing the pressure in the calorimeter.

It is most important that all parts of calorimeters of this type, as well as the connections leading to the main steam pipe, should be very thoroughly lagged by a covering of good insulating material. One of the best materials for this use is hair felt, and it is particularly well suited for covering the more or less temporary pipe fittings, valves, and nipples through which steam is brought to the calorimeter. Very many throttling calorimeters have been declared useless by engineers and put into the scrap heap merely because the small pipes leading to the calorimeters were not properly lagged, so that there was too much radiation, producing, of course, condensation, so that the calorimeter did not get a true sample. It is obvious that if the entering steam contains too much moisture the drying action due to the throttling in the orifice may not be sufficient

to superheat. It may be stated in general that unless there is about 5 to 10 degrees Fahrenheit of superheat in the calorimeter, or in other words unless the temperature on the low pressure side of the orifice is at least about 5 to 10 degrees Fahrenheit higher than that corresponding to the pressure **in**

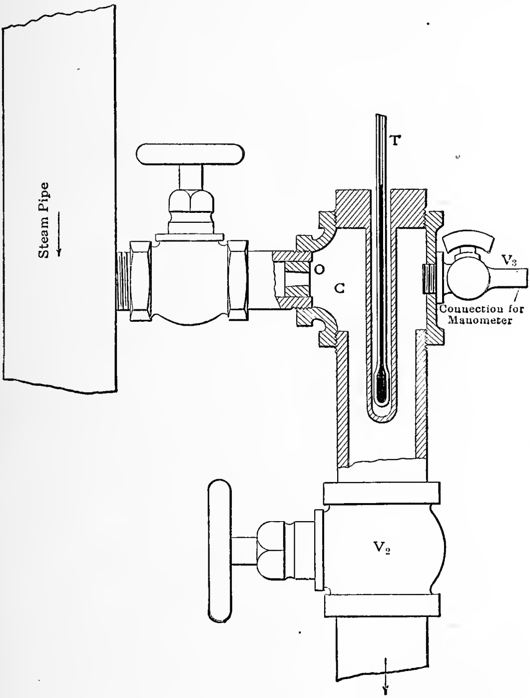


FIG. 52.—A Simple Throttling Steam Calorimeter.

the calorimeter, there may be some doubt as to the accuracy of results.[1] The working limits of throttling calorimeters vary

[1] The same general statement may be made as regards determinations of superheat in engine and turbine tests. Experience has shown that tests made with from 0 to 10 degrees Fahrenheit superheat are not reliable, and that the steam consumption in many cases is not con-

with the initial pressure of the steam. For 35 pounds per square inch absolute pressure the calorimeter ceases to superheat when the percentage of moisture exceeds about 2 per cent; for 150 pounds absolute pressure, when the moisture exceeds about 5 per cent; and for 250 pounds absolute pressure, when it is in excess of about 7 per cent. For any given pressure the exact limit varies slightly, however, with the pressure in the calorimeter.

In connection with a report on the standardizing of engine tests, the American Society of Mechanical Engineers[1] published the following instructions regarding the method to be used for obtaining a fair sample of steam from the main pipes. It is recommended in this report that the calorimeter shall be connected with as short intermediate piping as possible with a so-called **calorimeter nipple** made of $\frac{1}{2}$-inch pipe and long enough to extend into the steam pipe to within $\frac{1}{2}$ inch of the opposite wall. The end of this nipple is to be plugged so that the steam must enter through not less than twenty $\frac{1}{8}$-inch holes drilled around and along its length. None of these holes shall be less than $\frac{1}{2}$ inch from the inner side of the steam pipe. The sample of steam should always be taken from a vertical pipe as near as possible to the engine, turbine, or boiler being tested. Good examples of calorimeter nipples are illustrated in **Figs. 54** and **59.**

Never close and usually do not attempt to adjust the discharge valve $\mathbf{V_2}$ without first closing the gage cock, $\mathbf{V_3}$. Unless this precaution is taken the pressure may be suddenly increased in the chamber **C**, so that if a manometer is used the mercury will be blown out of it, and if, on the other hand, a low-pressure steam gage is used it may be ruined by exposing it to a pressure much beyond its scale.

Usually it is a safe rule to begin to take observations of temperature in calorimeters after the thermometer has indicated a maximum value and has again receded slightly from it.

sistent when compared with results obtained with wet or more highly superheated steam. The errors mentioned, when they occur, are probably due to the fact that in steam, indicating less than 10 degrees Fahrenheit superheat, water in the liquid state may be taken up in "slugs" and carried along without being entirely evaporated.

[1] *Proceedings American Society of Mechanical Engineers*, vol. 21.

The quality or relative dryness of wet steam is easily calculated by the following method. Using the symbols,

p_1 = steam pressure in main, lbs. per sq.in. abs.
p_2 = steam pressure in calorimeter, lbs. per sq. in. abs.
t_c = temperature in calorimeter deg. Fahr.
r_1 and q_1 = heat of vaporization, and heat of liquid corresponding to pressure p_1, B.T.U.
H_2 and t_2 = total heat (B.T.U.) and temperature (degs. Fahr.) corresponding to pressure p_2.
c_p = specific heat of superheated steam. Assume 0.5 for low pressures existing in calorimeters.[1]
x_1 = initial quality of steam, per cent.
$1 - x_1$ = initial moisture in steam, per cent.

Total heat in a pound of wet steam flowing into the orifice is

$$x_1 r_1 + q_1,$$

and after expansion assuming all the moisture is evaporated, the total heat of the same weight of steam is,

$$H_2 + c_p(t_c - t_2).$$

Then assuming no heat losses and putting for c_p its value 0.5 we have,

$$x_1 r_1 + q_1 = H_2 + 0.5(t_c - t_2), \quad . \quad . \quad . \quad . \quad . \quad (3)$$

or

$$x_1 = \frac{H_2 + 0.5(t_c - t_2) - q_1}{r_1}. \quad . \quad . \quad . \quad . \quad . \quad (3')$$

Chart for Moisture Determinations. Using degrees of superheat in the calorimeter as abscissas and initial absolute steam pressures as ordinates, the diagram in **Fig. 53** has been constructed. In the calculations it was assumed that the pressure in the calorimeter was atmospheric. In cases where this condition exists, therefore, after determining the degrees of superheat in the calorimeter ($t_c - 212$) and the initial absolute pressure p_1, the percentage moisture can be read from the curves without further calculations.

[1] Average values for the specific heat of superheated steam for any temperatures are given on page

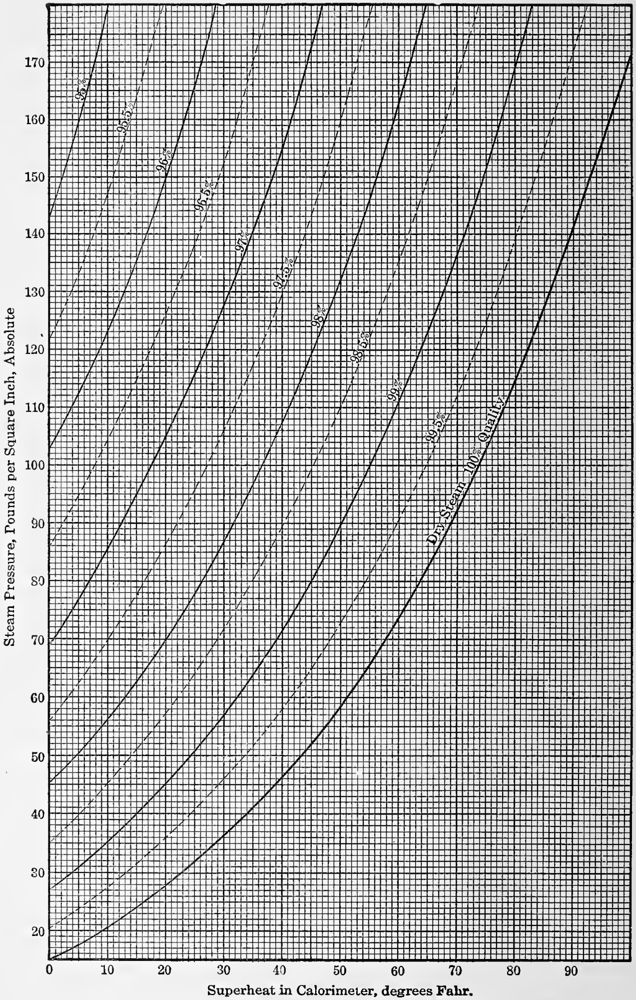


Fig. 53.—Chart for Determining Quality of Steam from Pressure and Superheat.

Although all the calculations for drawing this diagram were made by assuming atmospheric pressure in the calorimeter, the curves in the figure can be used with almost equal accuracy for pressures in the calorimeter not exceeding about 5 pounds above atmospheric by using the diagram[1] as if the ordinates represented the difference in pressure between the two sides of the orifice, or in other words the difference in pressure between that in the steam pipe leading to the calorimeter and that in the calorimeter itself. For example, if the superheat in the calorimeter is 40 degrees Fahrenheit, the initial steam pressure is 150 pounds per square inch gage and that in the calorimeter is 5 pounds per square inch gage, then the difference in pressure is 145 pounds per square inch, which is **equivalent,** with a barometric pressure of 15 pounds per square inch, to 160 pounds per square inch absolute pressure. Selecting the abscissa = 40 and the ordinate = 160 the diagram shows that the quality of the steam is approximately 96.9 per cent. If with the same data the quality $\mathbf{x}_1$ is calculated by equation (2), taking again the barometric pressure to be 15 pounds per square inch absolute, it is found to be 96.85 per cent, which for the accuracy required in the ordinary daily power plant calculations is good enough agreement, particularly when it is generally conceded by practical engineers that it is almost impossible with any of the **simpler** forms of steam sampling devices to obtain samples which do not differ as much as $\frac{1}{4}$ per cent from the average.

When a U-tube manometer is used to determine the pressure in a calorimeter of the type illustrated in **Fig. 52,** this pressure can be obtained very accurately and an excellent means is provided for **calibrating the thermometer** in the calorimeter just as it is to be used. The calibration would be made, of course, by the method of comparing with the temperature corresponding to known pressures explained on page 29. In order to avoid having superheated steam in the calorimeter for this calibration the felt or similar material usually needed for covering the valves and nipples between the main steam pipe and the calorimeter should be kept saturated with cold water.

[1] For the use of this diagram as well as the one in Fig. 55, acknowledgment is due to Messrs. Schaeffer & Budenberg and R. C. Carpenter.

The Barrus Throttling Calorimeter. An important variation from the type of throttling calorimeter shown in **Fig. 52** has been introduced quite widely by Mr. George H. Barrus. In this apparatus the **temperature** of the steam admitted to the calorimeter is observed instead of the pressure and a very free exhaust is provided so that the pressure in the calorimeter is atmospheric. This arrangement simplifies very much the observations to be taken, as the quality of the steam $\mathbf{x}_1$ can be calculated by equation (3′) by observing only the two temperatures $\mathbf{t}_1$ and $\mathbf{t}_2$, taken respectively on the high and low pressure sides of the orifice in the calorimeter. This calorimeter is illustrated in **Fig. 54.** The two thermometers required are shown in the figure. Arrows indicate the path of the steam.[1]

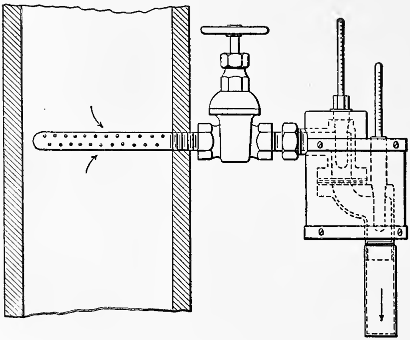


FIG. 54. Barrus Throttling Steam Calorimeter.

The orifice in such calorimeters is usually made about $\frac{3}{32}$ inch in diameter; and for this size of orifice the weight of steam[2] discharged per hour at 175 pounds per square inch absolute pressure is about 60 pounds. It is important that the orifice should always be kept clean, because if it becomes obstructed there will be a reduced quantity of steam passing through the instrument, making the error due to radiation relatively more important.

In order to free the orifice from dirt or other obstructions the connecting pipe to be used for attaching the calorimeter to the main steam pipe should be blown out thoroughly with

[1] *Proceedings American Society of Mechanical Engineers,* **vol.** 11, page 790.

[2] Formulas for calculating the exact weight of steam discharged from a nozzle are given on pages 148 and 149. In boiler-tests corrections should be made for the steam discharged from the steam calorimeters.

steam before the calorimeter is put in place. The connecting pipe and valve should be covered with hair felting not less than $\frac{3}{4}$ inch thick. It is desirable also that there should be no leak at any point about the apparatus, either in the stuffing-box of the supply valve, the pipe joints, or in the union.

Fig. 55 is a diagram for the determination of the quality of steam which is particularly suitable for use in connection with calorimeters of the Barrus type.

Abscissas in this diagram are temperatures in the calorimeter $\mathbf{t}_2$, and the ordinates are the initial temperatures $\mathbf{t}_1$ of the steam before expansion in the calorimeter.

With the help of such a diagram the Barrus calorimeter is particularly well suited for use in power plants, where the quality of the steam is entered regularly on the log sheets. The percentage of moisture is obtained immediately from two observations without any calculations.

Separating Calorimeters. It was explained on page 48 that throttling calorimeters cannot be used for the determination of the quality of steam when for comparatively low pressures the moisture is in excess of 2 per cent, and when for average boiler pressures in modern engineering practice it exceeds 5 per cent. For higher percentages of moisture than these low limits separating calorimeters are most generally used. In these instruments the water is removed from the sample of steam by mechanical separation just as it is done in the ordinary steam separator installed in the steam mains of a power plant. There is provided, of course, a device for determining while the calorimeter is in operation, usually by means of a calibrated gage glass, the amount of moisture collected. This mechanical separation depends for its action on changing very abruptly the direction of flow of wet steam moving with considerable velocity. Then since the moisture (water) is nearly 300 times as heavy as steam at the usual pressures delivered to the engine, the moisture will be deposited because of its greater inertia.

One of the simplest forms of separating calorimeters, made of pipe fittings, is shown in **Fig. 56.** Steam enters at **A,** passes down through the vertical pipe **P,** plugged at the lower end, from which it escapes through a large number of $\frac{1}{8}$-inch holes indicated in the figure. In passing through these holes the

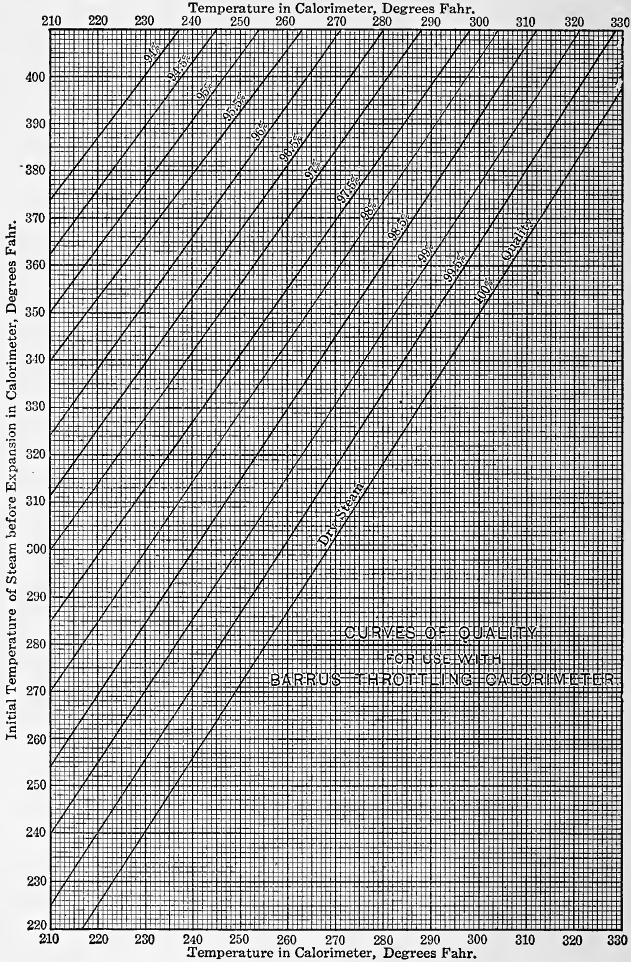


Fig. 55.—Chart for Determining Quality of Steam from Temperature Observations.

direction of flow is changed very abruptly, since the steam must go upward to be discharged at **D.** Moisture is deposited at the bottom of the vessel **V,** and its volume or weight can be determined from the height of the water in the gage glass **G** if the vessel has been calibrated. Steam discharged from **D** must be condensed and weighed in a pail or barrel containing cold water. The percentage of moisture is then found by dividing the weight of water collected in the vessel **V** by the sum of the weight of steam condensed and the weight of water collected in **V.** This sum is, of course, the weight of the **wet** steam.

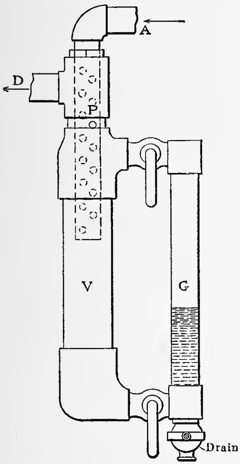


FIG. 56.—A Simple Separating Steam Calorimeter Made of Pipe Fittings.

Radiation Loss. As in all calorimetry work, in order to obtain accurate results there should be a covering of hair felt $\frac{3}{4}$ inch thick over all parts of the apparatus, and even then the radiation loss is sometimes large enough to make corrections necessary. This correction is determined by operating two calorimeters which are exactly alike in construction and in the amount of felt covering, in series, and so arranged that the second takes the discharge of the first. If it is known that the discharge from the first calorimeter is perfectly dry steam[1] then the moisture collected in the second calorimeter is the condensation due to its own radiating surface, which should be the same as for the first. Calculations for corrected moisture determinations are made then by subtracting from the moisture collected in the first instrument the amount condensed in the second. When, of course, the radiation loss has been once determined it is not necessary to operate the second calorimeter.

[1] A small throttling calorimeter can be attached to the discharge from the first separating calorimeter to determine whether or not the steam discharged is dry.

Carpenter's Separating Calorimeter. **Fig. 57** illustrates a form of separating calorimeter in which the improvement over the one shown in **Fig. 56** is in the addition of a steam jacketing space receiving live steam at the same temperature as the sample.

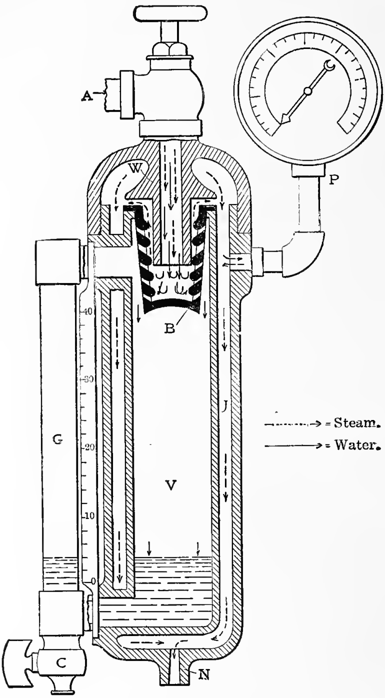


Fig. 57.—Carpenter Separating Steam Calorimeter.

Steam is supplied through a pipe **A,** discharging into a cup **B.** Here the direction of the flow is changed through nearly 180 degrees, causing the moisture to be thrown outward through the meshes in the cup into the vessel **V.** The dry steam passes upward through the spaces between the webs **W,** into the top of the outside jacketing chamber **J,** and is finally discharged from the bottom of this steam jacket through the nozzle **N.** This nozzle is considerably smaller than any other section through which the steam flows, so that there is no appreciable difference between the pressures in the calorimeter proper and the jacket. The scale opposite the gage glass **G** is graduated to show in **hundredths of a pound** at the temperature corresponding to steam at ordinary working pressures, the variation of the level of the water accumulating. A steam pressure gage **P** indicates the pressure in the jacket **J,** and since the

flow of steam through the nozzle **N** is roughly proportional to the pressure (see page 148), another scale in addition to the one reading pressures is provided at the outer edge of the dial. A petcock **C** is used for draining the water from the instrument, and by weighing the water collected corresponding to a given difference in the level in the gage **G**, the scale opposite it can be readily calibrated. Too much reliance should not be placed on the readings for the flow of steam as indicated by the gage, **P**, unless it is frequently calibrated. Usually it is very little trouble to connect a tube to the nozzle **N**, and condense the steam discharged in a large pail nearly filled with water. When a test for quality is to be made by this method the pail nearly filled with cold water is carefully weighed and then at the moment when the level of the water in the water gage **G** has been observed the tube attached to the nozzle, **N** is immediately placed under the surface of the water in the pail. The test should be stopped before the water gets so hot that some weight is lost by "steaming." The gage **P** is generally calibrated to read pounds of steam flowing in ten minutes. For the best accuracy it is desirable to use a pail with a tightly fitting cover into which a hole just the size of the tube has been cut

Combined Separating and Throttling Calorimeters. Calorimeters described are effective in removing practically all of the moisture in steam when the pressure is not lower than 25 pounds gage pressure. For lower pressures, particularly around atmospheric, recent experiments show that the efficiency of such carloimeters is in some cases not more than 80 per cent.[1] For this reason in the best current practice for determinations of moisture in low-pressure steam a throttling calorimeter is attached to the discharge of the separating calorimeter. Then if the separating calorimeter has been carefully calibrated for radiation loss and the steam escaping from the separating calorimeter is tested again in a throttling instrument, it is possible to make correct determinations for the percentage of moisture in the steam of almost any degree of wetness. An apparatus of this kind which is reported to have done excellent

[1] *Proceedings American Society of Mechanical Engineers*, Aug., 1910, page 1132. The efficiency of the calorimeter is the ratio of the percentage of moisture taken out by the separating calorimeter to the total percentage of moisture.

service in tests of very large low-pressure steam turbines, operating with the exhaust from reciprocating steam engines in New York city, is shown in **Fig. 58.** The most unique feature of this apparatus is the sampling tube. It was found that for this low-pressure steam the ordinary sampling tube of perforated pipe (see Figs. 54 and 59) did not give a reliable sample. It was also found necessary that the sample should be taken from the main

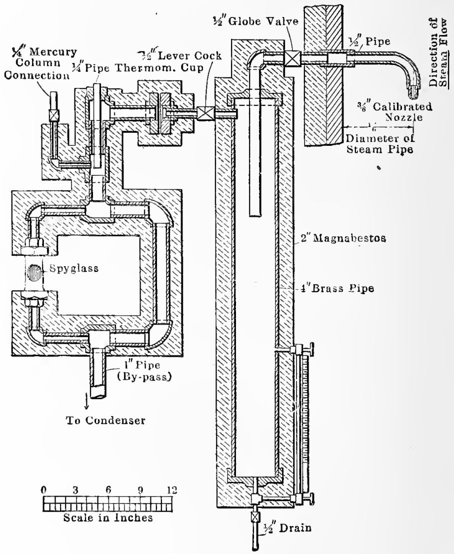


Fig. 58.—Stott's Combined Separating and Throttling Steam Calorimeter.

without changing its direction or velocity until it is actually inside the sampling pipe. If the direction of flow of wet steam is suddenly changed when entering the sampling nozzle, the entrained moisture, because of its greater specific gravity on the one hand and the very slight skin friction between it and the surrounding dry steam on the other, will cause it to continue in its path in a straight line, so that there is a tendency for only dry steam to enter the nozzle. Also if the velocity of the steam in

the sampler is greater than that in the main, there is a tendency for the dry steam to "accelerate" into the nozzle, leaving the moisture behind. It has been stated that this action has not been observed in tests of steam at high pressures, because (1) of smaller differences between the specific gravity of high pressure steam and water; (2) greater skin friction; (3) the highly divided state of the moisture.[1]

As the throttling calorimeter is ordinarily used it would have very little capacity when used with steam pressures only a little above atmospheric; but by making it discharge into a receiver in which a vacuum of 28 inches was maintained the throttling portion of the calorimeter will evaporate 2 to 3 per cent of moisture.

The apparatus shown in **Fig. 58** consists of the $\frac{3}{8}$-inch brass nozzle on the sampling tube which is bent to point in the direction opposite to that of the flow of the steam. The lip of this nozzle is filed to a knife-edge to avoid disturbing the current of steam around the mouth of the sampler by eddies and impact against a thick lip. This sampling tube is set up so that it extends into the main steam pipe $\frac{1}{6}$ of the diameter of the pipe, where it has been observed to give practically the true average flow. When in operation the valve at the sampling tube is opened wide and the flow is regulated by means of the lever cock between the separating and the throttling calorimeters. The necessary throttling action ordinarily produced by an orifice is produced by this cock. A vacuum is maintained in the throttling portion from which the discharge is carried to a small cooling receiver in which the steam is condensed. From this receiver it flows to a "volumetric" measuring tank of which the top is tightly closed and connected by $\frac{1}{4}$-inch pipes to the main condenser. A spy-glass shown at the left in the figure is useful for proving that the calorimeter is working properly. It often happens that when the superheat in the calorimeter is less than 5 to 8 degrees Fahrenheit there is some moisture passing through and the spy-glass will invariably show it. As the spy-glass is most conveniently made of $\frac{3}{4}$-inch gage glass its area is not large enough to carry all the steam, and a by-pass connection is arranged as shown. The large size

[1] H. G. Stott, *Proceedings American Society of Mechanical Engineers*, Aug., 1910.

of the parts is necessary on account of the very large specific volume of the low-pressure steam.

All parts of the apparatus are carefully covered with magnesia-asbestos covering 2 inches thick. For the normal rate of flow for the instrument, the radiation can be made less than 0.1 per cent.

Calculation of Percentage Moisture for Combination Separating and Throttling Calorimeter. Quality of steam x_1 is calculated for a combination calorimeter as follows:

Let w_1 = weight of moisture collected in the separating calorimeter in a given time, in pounds.

w_2 = weight of dry steam condensed after passing through the throttling calorimeter, in pounds.

x_2 = quality of steam discharged from separating portion as determined by the throttling calorimeter, then without sensible error the percentage of moisture in the steam is,

$$1 - x_1 = \frac{w_1}{w_1 + w_2} + x_2,$$

and in terms of "quality," we have also approximately,

$$x_1 = \frac{w_2}{w_1 + w_2} + x_2.\text{[1]} \qquad \ldots \ldots \quad (4)$$

Still another type of combined calorimeter is illustrated in **Fig. 59.** In this instrument, known as Ellison's **Combined Throttling and Separating Calorimeter,** the sample of steam is collected by the perforated tube in the main steam pipe. The temperature before expansion in the throttling plug is indicated by the thermometer marked T_1, and another thermometer T_2 gives the temperature after throttling. A scale **S** opposite the glass water gage **G** is used to show the weight of water separated from the steam. The proportion of moisture

[1] If the radiation test shows it is large enough to be appreciable, then if **R** is weight of condensation due to radiation in pounds in a given time corresponding to that for the other units, then

$$x_1 = \frac{w_2 + R}{w_1 + w_2} + x_2. \qquad \ldots \ldots \ldots \quad (4)$$

separated in relation to the weight of steam passing through the instrument is the percentage of moisture separated. This percentage is to be added to the percentage of moisture determined by throttling as calculated from the readings of the thermometers.

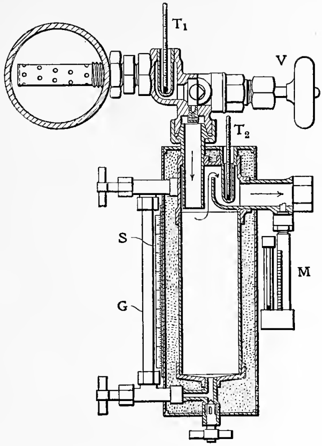


Fig. 59.—Ellison's Improved Steam Calorimeter.

Electric Steam Calorimeters. For use with particularly very wet steam, the Thomas electric calorimeter, **Fig. 60,** has been designed. It consists essentially of a cylindrical vessel **B** containing a series of resistance coils for heating steam by means of the electric current passing through them. These coils are connected to the electric terminals or binding-posts shown in the figure, and are supported in a soapstone cylinder in which there are a large number of $\frac{1}{4}$-inch holes through which the coils pass.

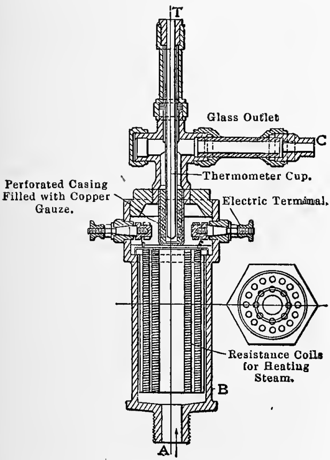


Fig. 60.—Thomas' Electrical Steam Calorimeter.

Steam enters at the bottom of the vessel at **A** and passing upward through the heated coils the moisture contained in it is evaporated. The steam then passes up through a perforated casing filled with copper gauze and escapes through the pipe discharg-

ing at the side at **C**. A part of this latter pipe is made of a glass tube for observing the condition of the steam. A thermometer is inserted at **T** for observing the temperature of the steam after this reheating.

Although this apparatus is used for steam of high quality as well as low, it has not been generally used to any great extent, probably because throttling calorimeters are preferred because of the greater simplicity and because very often a source of electric current is not conveniently available where tests are to be made. No data are available comparing its efficiency with that of the combined separating and throttling calorimeters described in the preceding paragraphs, but for **accurate** tests the latter are generally preferred by engineers.

Barrel Calorimeters. There is still another kind of steam calorimeter, known as the barrel type, deserving some attention. It is one of the oldest forms of apparatus for making determinations of the quality of steam. In the classification made at the beginning of this chapter it belongs in the group of **condensing** calorimeters. Even with expert manipulations, ordinarily it is much less accurate than any of the calorimeters already described. A typical apparatus of this kind is shown in **Fig. 61.** It consists usually of a weighing barrel **B,** made of three concentric vessels of galvanized iron with the two annular spaces between the inner and outer vessels filled with pressed sheet cork or hair felt to reduce radiation to a minimum. It is usually arranged so that when the inner vessel has been nearly filled with water from the barrel **A,** a quantity of the steam to be tested can be passed into it. The steam is admitted into the barrel in the most common forms by disconnecting the water hose **R** at **C** and making a temporary connection from the steam pipe out of which a sample is to be taken to a vertical pipe in the barrel of sufficient length to extend nearly to the bottom of the inner vessel. The pipe may be plugged at the lower end and sufficient area for the escape of steam is then secured by drilling into the pipe a number of $\frac{1}{8}$-inch holes for some distance from the lower end. This arrangement will make it easier to secure an equal rise in temperature in the different parts of the barrel. A float is usually provided to show the depth of water in the barrel, and a suitable stirring device or agitator consisting of paddles

attached to a vertical shaft is also needed. This agitator when revolved stirs up the water and brings it to a constant temperature.

Briefly the method to be pursued in the operation of the barrel calorimeter may be outlined as follows: First fill the barrel with cold water till the float shows that the water level is within about 6 inches from the top. Then stir well, observe

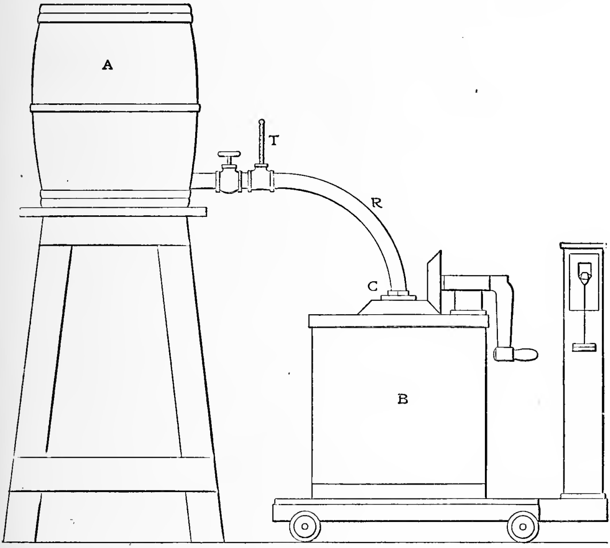


FIG. 61.—Barrel Steam Calorimeter.

the temperature accurately and weigh carefully on a platform scales. The steam pipe should then be connected up to discharge into the water after first allowing the steam to blow off into the air, for the purpose not only of removing the condensation in the piping, but also to heat it to as nearly as possible the temperature of the steam. When the temperature has risen to about 120 degrees Fahrenheit the steam should be shut off and another weighing made to determine the amount of steam added. While the weighing is being done the water should be

stirred vigorously and the highest temperature observed. For all the weighings the piping must be in exactly the same position as regards the connection on the barrel and all the pressure in the pipes must be relieved. When the piping between the calorimeter and the steam supply is connected by pipe fitters' unions these should be disconnected to insure the best accuracy. When, however, the connection is made by means of flexible rubber hose the weight can probably be obtained accurately enough without disconnecting the piping if the precaution is taken to relieve the pressure in the piping by opening a petcock located in the steam pipe near the point where it enters the barrel.

If, just before making the test for the quality of the steam, the calorimeter is filled with water, heated with steam or otherwise to about 150 degrees Fahrenheit and again carefully drained, the barrel will be near its average temperature during the test, and no correction need probably be made for the heat absorbed by the calorimeter. In most cases it is preferable, however, to determine accurately the heat absorbed by the calorimeter and then make the proper corrections; but unless the work be done very carefully it is valueless. This correction is usually made by calculating the **water equivalent** or the capacity of the calorimeter to absorb heat measured by the similar capacity of water. This water equivalent is to be added to the weight of water in the calorimeter. Using then the following symbols:

$\mathbf{w}'$ = weight of water in calorimeter in lbs.
$\mathbf{w}''$ = weight of water added in lbs.
$\mathbf{t}'$ = temperature of water in calorimeter, deg. Fahr.
$\mathbf{t}''$ = temperature of water added, deg. Fahr.
$\mathbf{t}'''$ = temperature of mixture, deg. Fahr.
$\mathbf{k}$ = water equivalent in lbs. Then

$$(\mathrm{w}'+\mathrm{k})(\mathrm{t}'''-\mathrm{t}')=\mathrm{w}''(\mathrm{t}''-\mathrm{t}'''),$$

$$\mathrm{k}=\frac{\mathrm{w}''(\mathrm{t}''-\mathrm{t}''')}{\mathrm{t}'''-\mathrm{t}'}-\mathrm{w}', \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

The temperature of the water added should be taken just as it **enters** the calorimeter and as near to it as possible.

The quality of the steam ($\mathbf{x}_0$) to be determined is calculated as follows:

w_1 = weight of water in calorimeter in lbs.
w_2 = weight of steam added, in lbs.
k = water equivalent of calorimeter in lbs.
t_1 = initial temperature of water in calorimeter, degs. Fahr.
t_2 = final temperature of water, degs. Fah.
p_0 = pressure of steam, lbs. per sq.in.
r_0 = heat of vaporization of steam (B.T.U.) corresponding to p_0.
q_0 and q_2 = sensible heat of steam (B.T.U.) corresponding to p_0 and t_2.

Then equating the heat lost by the steam to the heat gained by the water[1],

$$(x_0r_0 + q_0 - q_2)w_2 = (w_1 + k)(t_2 - t_1). \quad . \quad . \quad . \quad (6)$$

$$x_0 = \frac{(w_1 + k)(t_2 - t_1)}{w_2r_0} + \frac{q_2 - q_0}{r_0}. \quad . \quad . \quad . \quad . \quad (7)$$

The accuracy of this instrument depends principally on the accuracy with which the various temperatures and the weight of the condensed steam are obtained. Usually it is very difficult to obtain accurately the temperature of the mixtures of water and steam. It is not unusual for determinations of moisture with such a calorimeter to vary for the same quality of steam and with expert handling as much as 5 per cent. In the case, therefore, of steam with 10 per cent moisture, the determination of quality might be in error as much as one-half per cent.

Calorimeter Calibrations. For a laboratory calibration exercise three calorimeters of different types are connected by means of exactly the same kind of fittings and valves to the same steam main or receiver. A water-jacket or a device like that shown in **Fig. 34** should be provided to vary the quality of the steam. Tests should be made simultaneously and for the same length of time on the three instruments.

[1] Equation (7) can be made a little simpler for calculations by writing for $(q_2 - q_0)$ the difference between the corresponding temperatures $(t_2 - t_0)$ without appreciable error.

CHAPTER IV

MEASUREMENT OF AREAS

Planimeters. The most accurate and generally approved method for obtaining the area of irregular figures is by means of integrating instruments called planimeters. Instruments of this kind may differ in many details, yet all of them are based, in theory, on the original Amsler **polar planimeter.**

Polar Planimeters. One of the simplest forms of the polar type of planimeters is shown in **Fig. 62.** It consists essentially

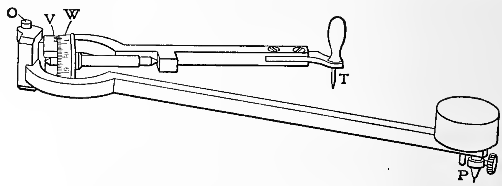


FIG. 62.—Amsler Polar Planimeter.

of two arms **PO** and **TO** pivoted together at **O.** When in use the point **P** is not to be moved, and is held in place by means of a pin-point upon which a small weight rests. There is a **tracing point** at **T** intended to be moved around the border of the area to be measured. Attached to the arm **TO** is a small graduated wheel **W** carried on a short axis which must be placed accurately parallel to **TO.** Any movement of the arm **TO** except in the direction of its axis will, of course, move the wheel **W** on the paper or other surface on which it is placed so that the amount of its movement gives a record indicating the area measured. A vernier, **V,** placed opposite the graduations on the wheel, assists in reading the instrument accurately. The arm **TO** is usually made of such a length that the movement of the tracing point **T** around an area of one square inch (for English

units) will move the wheel one-tenth of its circumference. Graduations of the vernier indicate usually one one-thousandth of a revolution of the wheel, or in English units one one-hundredth of a square inch.

When the tracing point T is moved around an area in a clockwise direction the wheel will roll in the direction of its graduation, and the area is found by subtracting the final reading from the initial. Amsler planimeters are often constructed with the arm **OT** adjustable in length, so that it can be set to indicate areas in various units, as, for example, square inches, square feet, square centimeters, etc.

The veriner **V** has ten graduations, and the total length of these ten divisions is one-tenth less than the length of those on the wheel, so that it represents, counted from zero, so many hundredths of an inch. To explain the method of using the vernier, **Fig. 63** has been inserted, showing the wheel **W** and the vernier **V**, in a drawing of larger scale than in **Fig. 62.** Readings of the graduations on the wheel **W** are always taken opposite the zero mark on the vernier, so that the reading indicated in **Fig. 63** without the help of the vernier would be little more than 4.7. The graduation on the vernier which is exactly coincident with a graduation on the roller wheel is the third from zero and indicates three hundredths. The complete reading is therefore 4.73 as determined by the vernier.

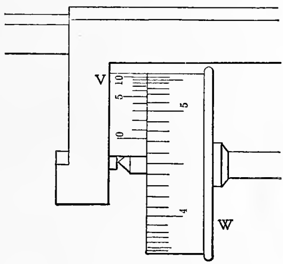


FIG. 63.—Typical Vernier for a Planimeter.

Theory of Polar Planimeters. As this instrument is constructed neither of the points **T** nor **W** can **pass over** the arm **PO** (**Fig.** 64). If the arms **PO** and **TO** are clamped so that the plane of the graduated wheel **W** intersects the point **P**; that is, when the angle **TWP** is a right angle, and then the arms thus clamped are revolved around this point, the wheel will be continually slipping without any rolling motion in the direction of its axis, and consequently it will not revolve. When, however, the arms are not clamped and if the con-

struction of the instrument will permit the tracing point **T** to be moved out so far that the axis of **W** will lie in the line **PO**, then an arc described by the movement of **T** will produce only a **rolling** motion of the wheel. Obviously with the arms in any position intermediate between that of the clamped right angle and the one with **W** in line with **PO**, the wheel will partly slip and partly roll, the amount of slipping and rolling depending on the size of the angle between the arms. It follows, then, that when circumscribing a closed figure, the radial components cause only slipping of the wheel and need not be considered, while the circumferential components produce a resultant rolling which must be taken into consideration.

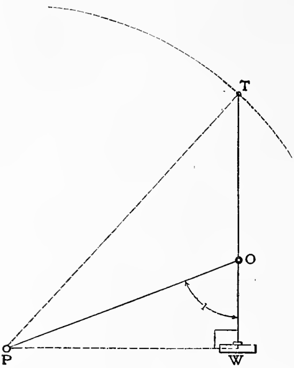


Fig. 64.—Position of the Arms of a Polar Planimeter to Draw the "Zero" Circle.

The path described by the tracing point **T** when the arms are clamped as indicated in **Fig. 64**, is called the **zero circle** for the planimeter. If the tracing point is moved in any path outside the zero circle in a clockwise direction a positive record will be indicated on the graduated wheel, while if it is moved in a path in the same direction as before but inside the zero circle, there will be a negative record.

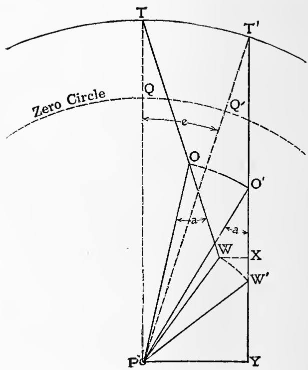


Fig. 65.—Theoretical Diagram for a Polar Planimeter.

According to the theory of polar planimeters, they are

designed so that the rolling of the wheel for a given circumferential motion of the tracing point **T** is proportional to the area included between the path of **T**, the radial line from **P** (Fig. 65) to the initial and final points of the path taken by **T**, and the arc of the zero circle included between these radial lines. In the discussion of this theory, circumferential motion of the tracing point **T** around the point **P**, and the angle **WOP** (marked **a**) remaining always at a constant value, is to be taken up first.[1] Now let us suppose the tracing-point is moved from **T** to **T′** in the figure through a very small angle, **TPT′** (marked **e**), keeping, however, the angle **a** constant, then the graduated wheel **W** will move through the arc **WW′**, partly rolling and partly slipping. The component of this motion producing **rolling** will be perpendicular to the axis of the wheel; or, in other words, this component will be perpendicular to **OT** in all its positions, and without appreciable error for small values it may be represented **in this figure** by the line **WX**, making **WXW′** a right-angled triangle of infinitesimal proportions. When the tracing point has moved from **T** to **T′** the point **O** has moved through the arc **OO′** and the tracing point subtends in its movement an angle **WPW′**, which is equal to the angle **TPT′**, marked **e**, which was passed over by **T**. Then the following relation is easily obtained:

$$\mathbf{WW'} = \mathbf{PW'} \times \mathbf{c}.$$

The symbol **c** is constant, expressing the ratio for a given angle **WPW′** between the length of an arc and the corresponding radius for any value of this radius. In other words in terms of the calculus this constant would be expressed in **radians**. In general for every angle there is a constant value which when multiplied by the radius gives the length of the arc for that radius.

The component of **WW′** corresponding to the **rolling** of the wheel is **WX**, which is approximately equal to the arc **WW′** times **cos W′WX**. That is,

$$\mathbf{WX} = \mathbf{PW'} \times \mathbf{c} \times \cos \mathbf{W'WX} \quad . \quad . \quad . \quad . \quad (8)$$

But if **PY** is drawn perpendicular to **T′W′** produced

$$\mathbf{PW'} \cos \mathbf{W'WX} = \mathbf{W'Y}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

[1] In the mathematical discussion following, the graduated wheel will be considered as if it were a part of the arm **TO**, with its plane exactly at right angles to the axis of this arm.

and combining (8) and (9),

$$W'Y = \frac{WX}{c}. \qquad (10)$$

Since the angle **WPW′** is very small, **WW′** may be taken as being perpendicular to **W′P**. Now **WX** is perpendicular to **T′Y** and the angle **W′WX** is equal to the angle **PW′Y.** The trigonometric relations reducing the above to terms of the length of one of the arms of the planimeter and the constant angle **a** are as follows:

$$W'Y = \frac{WX}{c} = PW' \cos PW'Y = PO' \cos PO'Y - W'O'$$

$$= PO' \cos a - W'O',$$

then $$WX = c(PO' \cos a - W'O'). \qquad (11)$$

This is an expression for the amount of **rolling** of the wheel when the tracing-point moves from **T** to **T′**.

To express the relations required the area will now be expressed trigonometrically in similar units. From geometry the area of sector $TPT' = 1/2$ arc $TT' \times PT$, but arc $TT' = PT \times c$ (see page 69), or **area** $TPT' = 1/2\ c \times \overline{PT}^2$.

We can write also,

$$PT = \sqrt{\overline{PO}^2 + \overline{OT}^2 + 2PO \times OT \cos a},$$

$$\text{area } TPT' = 1/2\ c\ (\overline{PO}^2 + \overline{OT}^2 + 2PO \times OT \cos a). \qquad (12)$$

But the area represented by the amount of rolling of the graduated wheel is that part of the sector **outside** the zero circle (see page 71), and this is the area **TT′Q′Q**. Now the radius **r** of the zero-circle, referring again to **Fig. 64***, is easily obtained from equations expressing the relations of the sides of the right triangles in that figure for the particular case when there can be no rolling movement. Thus,

$$\overline{PO}^2 = \overline{WO}^2 + \overline{PW}^2, \qquad (13)$$

$$\overline{PW}^2 = \overline{PT}^2 - \overline{WT}^2 = \overline{PT}^2 - \overline{WO}^2 - 2WO \times OT - \overline{OT}^2. \qquad (14)$$

* It will be remembered that with the arms of the planimeter in the position shown in **Fig. 64** the tracing point **T** describes the circumference of the zero circle.

Combining equations (13) and (14),

$$\overline{PO}^2 = \overline{WO}^2 + \overline{PT}^2 - \overline{WO}^2 - 2WO \times OT - \overline{OT}^2.$$

But $PT = r$, the radius of zero-circle, therefore,

$$r = \sqrt{\overline{PO}^2 + 2WO \times OT + \overline{OT}^2}. \quad . \quad . \quad . \quad . \quad . \qquad (15)$$

Also from geometry, as explained on the preceding page,

$$\text{Area } QPQ' = 1/2\ r \times c \times r = 1/2\ c \times r^2$$
$$= 1/2\ c(\overline{PO}^2 + 2WO \times OT + \overline{OT}^2). \qquad (16)$$

Subtracting equation (16) from equation (12).

$$\text{Area } QTT'Q' = c \times OT(PO \cos a - WO). \quad . \quad . \qquad (17)$$

Equation (17), which is the expression for the area outside the zero-circle, will be observed to be equivalent to the roll of the graduated wheel as given in equation (11), times the length of the arm **OT** from the pivot to the tracing-point. If, therefore, for a given area **A**, we call the reading of the wheel **R** and the length of the arm from pivot to tracing-point **L**, then,

$$A = LR. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (18)$$

It should be noted further that this equation is independent of any other dimensions of the instrument.

That this demonstration applies to areas not adjacent to the zero-circle or partly inside and out can be readily shown by subtracting in a given case the area between the zero-circle and the required area.

Area of Zero Circle by Experiment. The area of the zero-circle of a planimeter may be found readily by passing the tracing-point around the circumference of two circles each larger than the zero-circle. Preferably for this operation the fixed point of the instrument is placed at the center of the circles. If the calculated areas of these circles are respectively A_1 and A_2, and r is the radius of the zero-circle, then since readings of the graduated wheel show only the areas outside the zero-circle represented by R_1 and R_2, we obtain,

$$A_1 = \pi r^2 + R_1,$$

$$A_2 = \pi r^2 + R_2,$$

$$2\pi r^2 = A_1 + A_2 - (R_1 + R_2). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (19)$$

After **r** has been found [1] it is not difficult to calculate the proper length of thearm **OT** for any linear units (compare equation 15). In fact very many polar planimeters are constructed with an adjustable arm **OT,** so that the instrument can be used for any scale or for various units. The exact lengths required for both the English and metric units (inches and centimeters) are usually stamped on the adjustable arm.

Mean Ordinate of an Area.[2] If we call **m** the mean ordinate and **l** the length of a given area **A,** then

$$A = ml.$$

From equation (18) we have,

$$A = LR,$$

whence

$$ml = LR,$$

$$m = \frac{L}{l} R. \qquad (20)$$

When, therefore, the tracing-point arm is adjustable it may be set as shown in **Fig.** 66 [3] to make it equal to the length of

[1] If instead of measuring and calculating the circles both larger than the zero-circle, one of the two is made smaller than the zero-circle, then the reading of the instrument is again the difference between the area of the circle and that of the zero-circle, but the value of this difference is now negative, so that if A_1 is the area of the circle larger than the zero-circle and A_2 is the area of the one smaller, then using the other symbols as before,

$$A_1 = \pi r^2 + R_1,$$
$$A_2 = \pi r^2 - R_2,$$
$$2\pi r^2 = A_1 + A_2 - (R_1 - R_2).$$

Although this latter method does not fall in with the general demonstration so well, it is, however, usually preferred, as it will give greater accuracy than can be obtained with two circles both larger than the zero-circle, unless one of these is made unusually large.

[2] Engineers must calculate **mean ordinates** most often when determining the **mean effective pressure** (M.E.P.) of engine indicator diagrams.

[3] To facilitate the adjustment of the arm to the length of the diagram or area measured, sharp points **M** and **N** are attached to the back of some planimeters. The point **M** is often conveniently placed a short distance away from the tracing-point **T,** and the point **N** must then be the same distance and in the same direction away from the pivot **O.** Then obviously the distance between **M** and **N** will be in all cases equal to the length of the adjustable arm.

the area measured. Then, obviously, the height of the mean ordinate will be equal to the reading of the graduated wheel expressed in the same units. For example, if the subdivisions of the wheel are fortieths of an inch, the result will be the mean ordinate also in fortieths. This scale of the wheel is not determined by the diameter of the portion of the wheel which is graduated, but by the diameter of the **edge** which comes into contact with the surface over which the wheel rolls. If then **d** is the so-called diameter of "rolling" of the wheel, its circumference is π**d**. Now by dividing the number of divisions on the circumference (usually 100) by π**d**, the "scale"

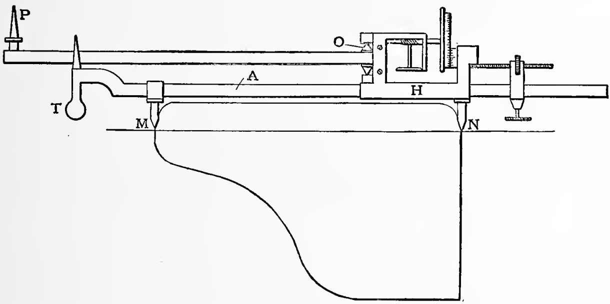


FIG. 66.—Polar Planimeter with Adjustable Arms for the Rapid Determination of Mean Ordinates.

of the wheel is obtained. It may also be found by measuring a rectangular area of the same length as that of the **tracer** arm and one inch wide, when the reading from the wheel will give the number of divisions per inch. For those instruments of which the radius of the wheel is one centimeter (.795 inch diameter) and having 100 divisions, the scale is almost exactly 40 divisions to the inch.

Coffin Planimeter and Averaging Instrument. This planimeter is made commonly in two forms, illustrated by **Figs. 67** and **68.** As regards details the former is somewhat the simpler and will be explained first. In principle the two are exactly alike. As will be observed in the figures, this instrument has a single arm to which a suitably graduated wheel is attached

on an axis parallel to the line joining the ends of the arm. One of the ends of this arm is for tracing the outline of the area measured while the other slides up and down in a suitable slot. One of the advantages of this instrument over the polar planimeter, although it is not so generally adaptable, is that the wheel is made to move over a specially prepared surface, preventing unnecessary slipping. On materials having a rough, fibrous, or worst of all, an uneven surface, the movement of the wheel of any planimeter will not be the same as when rolling over a smooth flat surface.

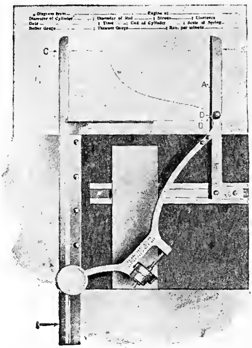


FIG. 67.—Coffin Planimeter.

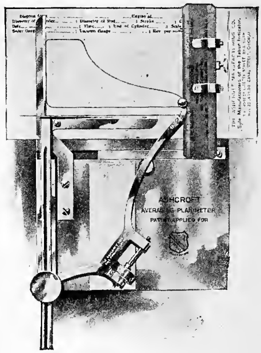


FIG. 68.—Coffin-Ashcroft Averaging Planimeter.

The Coffin planimeter may be discussed as a special form of the general **polar** type in which the pivoting point **O,** instead of **swinging** about the fixed point **P** (**Fig. 62**) moves back and forth in a straight line. The angle between the arms **PO** and **OT,** as indicated by the dotted lines in **Fig. 69,** is really invariable at 90 degrees. Obviously, then, the equation (17) expressing the area traced by a polar planimeter outside the zero circle becomes, referring to **Fig. 65,**

$$\mathbf{area} = \mathbf{c} \times \mathbf{OT}(-\mathbf{WO}),$$

likewise equation (11), expressing the roll of the wheel for the Coffin planimeter, becomes equivalent to,

$$\textbf{Roll or record of the wheel} = \mathbf{c}(-\mathbf{W'O'}) = \mathbf{c}(-\mathbf{WO}).$$

Using, as before, in equation (18), the symbols **L** and **R** for, respectively, the length of the arm **OT** and the reading of the wheel, we have, just as for the polar planimeter,

$$A = LR. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (21)$$

As an **averaging instrument** the Coffin planimeter is very much more convenient than the typical forms of polar planimeters. For finding the **mean ordinate** of an area the use of the polar type of these instruments was explained on page 72. The sliding vertical straight edge shown at the right in **Figs. 67** and **68** is

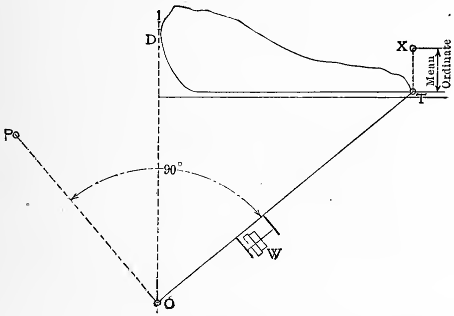


FIG. 69.—Theoretical Diagram for a Coffin Planimeter.

for the purpose of making the operation of finding the mean ordinate of an area (or the "mean effective pressure" of an engine indicator diagram) as simple as possible. For this operation the straight edges **C** and **K** should be adjusted so that when the tracing pin passes over the extreme end of the area to be measured it will just touch both of them. Now if the tracer is started at either **end** of the area and moved around to the starting point and then moved upward along the vertical straight edge until the reading of the wheel is the same as when starting to trace the area, this last distance traced from the starting point along the vertical straight edge is the **mean ordinate.** To demonstrate this statement the symbols used on page 72 will be continued. Representing the mean ordinate by **m,** the length of the area **A** by **l,** the reading or rolling of the gradu-

ated wheel in **going around the area** by **R,** and the length of the arm carrying the tracer by **L,** then as before

$$A = ml.$$

Now, when the tracing-point **T** moves over a vertical line, the angle **DOT,** represented by **Z** in **Fig. 70,** remains constant. If we call the vertical distance moved **v,** and remember that only the movement of the wheel at right angles to its axis produces rolling, then the reading corresponding to the rolling, **R,** is,

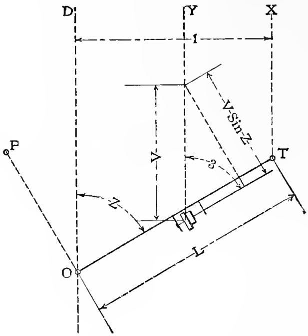


FIG. 70.—Theoretical Diagram for a Coffin Planimeter.

$$R = v \sin z. \quad . \quad . \quad . \quad (22)$$

But for the position shown in **Fig. 69** when the tracer **T** is at the right-hand end of the outline of the area, we have

$$\sin z = \frac{l}{L},$$

whence

$$R = \frac{vl}{L}.$$

Substituting this value of **R** in the general equation **(20)** for the mean ordinate **m** of a polar planimeter, then,

$$m = \frac{L}{l}\,\frac{vl}{L} = v. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (23)$$

This relation can be illustrated more simply, however, by referring to **Fig. 71,** which is typical indicator diagram from a steam engine. In this figure the tracing point of the Coffin instrument is shown at **O,** with the tracing arm represented by **VO.** A rectangle **OXYZ,** indicated by dotted lines, is shown of which the area is equal to that of the indicator diagram.

Starting at **O** and moving the tracing point around the **indicator diagram** once the difference in readings is the area. Now if the tracing point is moved in the opposite direction around the **rectangle** and again back to the starting point at **O** it will measure a negative area equal to the first area and the reading of the graduated wheel will be the same as when first started around the indicator diagram. The movement of the graduated wheel as the tracing point moves from **X** to **Y** is equal and opposite to that in going from **Z** to **O,** so that these two cancel each other. The motion of the tracing point from **Y** to **Z** requires the axis of the graduated wheel to be parallel to **YV** and consequently during this movement the wheel will not be moved. The only movement that is therefore producing a net change in reading of graduated wheel during the reverse tracing of the rectangle is in going from **O** to **X.** Consequently after going around any irregular area like an indicator diagram in a clockwise direction from the starting point at **O** at the right-hand end of diagram, if the tracing point is moved in a vertical direction from the starting point at **O** until the reading of the graduated wheel is the same as when first started, this vertical distance moved, measured from **O,** will be equal to the mean height of the indicator diagram.

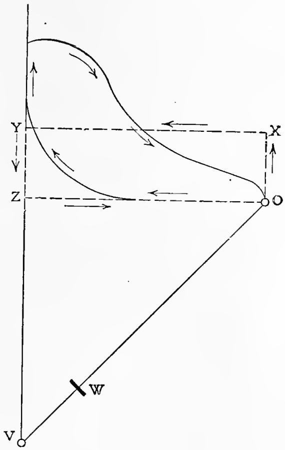


FIG. 71.—Diagram Explaining the Method of Mean Ordinates with a Coffin Planimeter.

Although measurements of areas may be made with the Coffin planimeter as with the regular polar types with the area in any position as regards its length and breadth, yet when the mean ordinate is to be obtained, its value in a definite position is required and the area must be placed so that its length with respect to which the mean ordinate is to be obtained will lie along the horizontal straight edge shown in the figures. Then

the mean ordinate measured along a vertical straight edge will give the result required.

Roller Planimeters. For the measuring of very large areas a planimeter differing slightly in theory from the polar type has been designed by G. Coradi, of Zurich, Switzerland. It has the advantage of being adaptable for measuring surfaces of indefinite length and as wide as the length of the tracer arm. This instrument is illustrated in **Fig. 72.** This instrument is supported at three points—the two rollers **R**^{1} and **R**^{2} and the tracing pin **f,** or its support **s.** These two rollers are attached

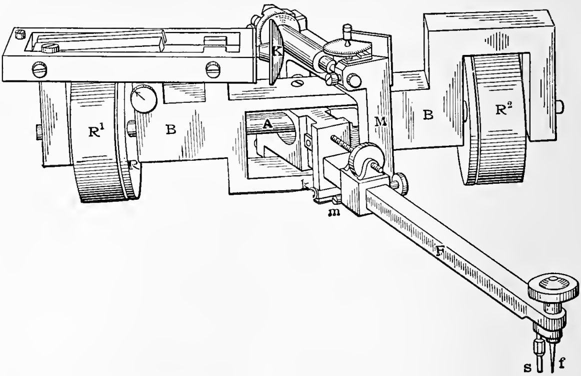


Fig. 72.—A Typical Roller Planimeter.

to the shaft **A.** On the face of one of these rollers is a minutely divided miter-wheel engaging with a small pinion revolving the horizontal shaft carrying the spherical segment **K.** At the center of the frame **B,** and in the same vertical plane with the two shafts already mentioned, a vertical shaft carrying the tracer arm is supported. The spherical segment **K** causes merely by friction contact the movement of the cylindrical "measuring" roller shown at its right. This roller is supported on the auxiliary frame **M,** of which the tracer arm is a part. The "measuring" roller moves back and forth with respect to the spherical segment to correspond with the movement of the tracing point; but at the same time

the rotation of the segment itself imparts rolling motion of the entire instrument.[1]

Testing Planimeters. Tests are made by comparing the readings of the instrument with that calculated for a given area. For such calibrations it is necessary to use an area which can be gone over accurately with the tracing point preferably held mechanically. This is done usually by using a metallic **testing rule** shown in **Fig. 73.** It is usually made in the shape of a narrow strip from three to five inches long. At the end marked zero on the graduations a needle point is set which is kept in place by an overlapping screw. At each line of the graduations there is a very small conical hole into which the tracing point of the planimeter can be placed. The beveled end of the testing rule has the index line to be set accurately at the starting mark, so that this point can be very carefully located. With the tracing-point **T** in the testing rule and the fixed point **P** of the planimeter in approximately the position shown in **Fig. 74,** observe the reading of the instrument corresponding to the area of the circle described by the tracer moving clockwise, **in the positions shown.**

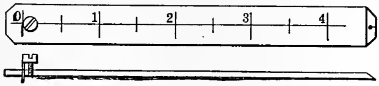


FIG. 73.—Planimeter Testing Rule.

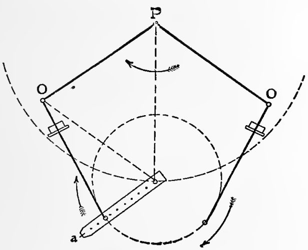


FIG. 74.—Methods of Testing Planimeters.

1st. When the fixed point **P** is on the left-hand side of the tracing point.

2d. When **P** is on the right-hand side.

If the reading obtained is greater in the **first** position than in the **second,** the end of the shaft carrying the graduated wheel

[1] Since this instrument is not often used by engineers, those interested in its theory are referred to Coradi's book of directions (in English) accompanying each instrument, or to *Handbuch der Vermessungskunde,* by W. Caville.

nearest the tracing point must be shifted toward the right to make the instrument accurate, and vice versa. Otherwise the error, if there be one, can be eliminated by taking the mean of the results obtained for the two positions[1].

Another test to be made, if there is doubt about the accuracy of a planimeter after the axis of the wheel has been adjusted, is to determine whether the settings of the adjustable arm marked on the instrument are correct. For this determination circles with several different diameters can be measured with the testing rule, and if there is a nearly constant **percentage** error, say **x** per cent too large, then the adjustable arm must be lengthened **x** per cent to make the planimeter readings correct, and vice versa.

For accurate results the fixed point **P** should be placed as indicated by the dotted lines in **Fig. 74,** so that when the tracing point is near the center of the area to be measured the two arms will be approximately at right angles.

Durand-Bristol Integrating Instrument. This instrument, illustrated in **Fig. 75,** has been recently developed by the Bristol Company for obtaining the average radius of records traced on circular charts of uniform graduations like those used in recording gages and thermometers, etc. It is a simple device for obtaining quickly the average value of pressure, temperature, draft, watts, volts, amperes and other records generally taken on circular charts.

This instrument consists of a wooden base in which there is a metal socket for supporting a rotatable pin slotted for receiving a horizontal shaft to which the integrating wheel is rigidly attached. On this shaft between the integrating wheel and the pin there is an adjustable tracing point and at the opposite end of the shaft there is a triangular support for the shaft, also adjustable.

The general principle of this instrument is due to Professor W. F. Durand[2] of Leland Stanford University. Its application

[1] For ordinary requirements a **testing disk** can be used in place of the rule, although it is not usually so accurate. On this disk circles of 1, 2, and 2½ inches diameter are usually engraved; and if neither a testing plate nor a disk is available, rough tests can be made by using circles drawn with a pencil compass on a flat sheet of well-calendered paper.

[2] *Proceedings American Society of Mechanical Engineers*, October, 1908, pages 1241–1246.

hinges on the condition that the chart to be measured has a uniform radial scale, the same as there must be a uniform vertical scale for indicator and other similar diagrams in order that they can be averaged with the ordinary planimeters. Obviously the mean value of the radius of a circular diagram cannot be determined with ordinary planimeters, since the area of a diagram in polar co-ordinates is proportional to the square

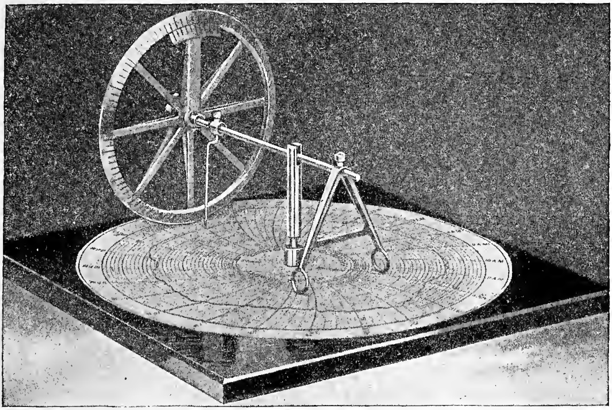


FIG. 75.—Bristol-Durand Integrating Instrument for Circular Charts.

of the radius and to the angle.[1] In **Fig. 76, AB** is an irregular curve, considered for this theoretical discussion as traced by a point moving in and out on a **straight radial** line. The center of the chart is at **O,** and at this point there is a socket, in which a rod **O′P** slides freely back and forth, permitting a tracing point, **P,** to draw a curve **AB.** A graduated wheel, **W,** attached to **O′P,** serves the same general purpose as the integrating wheel in the ordinary planimeter. Obviously this wheel will be moved only by circumferential motion, and for any radial movement of the rod in the direction of its length it will remain stationary.

[1] With the ordinary planimeter the mean square of the radial ordinates can be determined, and we can, of course, take the square root of these values, but in most cases this is not the same as the mean radius.

The amount of movement will be proportional to the radius **WO,** which differs from **PO** by a constant distance **PW.** Resultant movement of the wheel **W** is proportional, therefore, to the angle moved through by the arm **O′P** and to the radius **OW** varying from point to point along the curve. Assuming for the present, but as will be shown later the reading for any pare of the curve, as **AB,** to be proportional to the product of tht angle subtended between the points **A** and **B,** **AOB,** and the mean radius for the curve between these points, then if this reading is divided by the subtended angle expressed

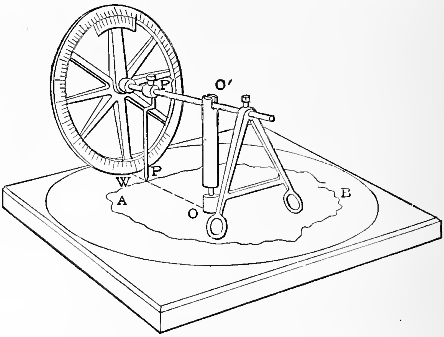


Fig. 76.—Diagrammatic Drawing of Bristol-Durand Integrating Instrument.

in circular measure the quotient will be proportional to the mean radius. Now if to the value of this mean radius the constant distance **WP** is added, the true value of the radial ordinate **OP** is obtained. When, as is usually the case in practice, the curve **AB,** represents values of radial ordinates with reference to à base circle of constant radius as the datum or "zero line," then if the radius of this base circle is subtracted from **OP** the remainder will be the true value of the ordinate. By making **WP** equal to the radius of the base circle, as may readily be done by a suitable adjustment of the instrument, the two corrections will be "balanced" and the mean value of the radial ordinate will be given directly as the quotient of the reading of the wheel and the subtended angle **AOB** expressed in circular measure. For a chart corresponding to

twenty-four hours for a circumference, the angular measure to be used as the divisor will be .2618 per hour.

The quantity to be determined in such diagrams is the time mean of the quantity measured by the radial ordinate. But since angular motion is made proportional to time, we may represent the desired mean by the following integral formula:

$$r = \frac{\int r\,d\theta}{\int d\theta} = \frac{\int r\,d\theta}{\theta} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (24)$$

Now, in **Fig. 77**, let **ABCD** denote a curve drawn by a tracing point which moves on the arc of a curve shown by **OAV** instead of on a straight radial line. Then let **OV, ON, OM,** etc., denote a series of consecutive positions of the curve **OAV,** at differential angular intervals **d**θ. Then for the actual curved path **ABCD** substitute the broken line path made up of a series of arcs each **r**$d\theta$ in length, and the series of differential bits of the curve **OAV** as shown. Then **at the limit** the record of any integrating or averaging instrument will be the same, whether the tracing point is carried along the curve or along the broken line substitute as shown.

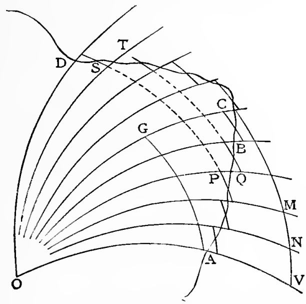


FIG. 77.—Theoretical Curves for Bristol-Durand Instrument.

Then suppose an integrating instrument, as shown in **Figs. 75** and **76,** is applied to such a diagram, and let the tracing point **P** be carried along the zig-zag path. The record of the wheel will be made up of two parts:

1. That due to the circular arcs **r**$d\theta$ and representing by summation the value of $\int r\,d\theta$.

2. That due to the differential portions of the arc **OAV.**

Now it is clear that if the diagram extends all the way around from **A** through **BCD** to **A** again the differential elements of the

curve **OAV** may be considered as existing in pairs, and that for every element traversed in the **outward** direction, there will be an equal element traversed in the **inward** direction. **PQ** and **ST** denote the members of such a pair. The record for such a pair will therefore disappear in the summation, and hence for all the pairs, and hence for the diagram as a whole. In such a case, therefore, part "2" above becomes zero and the record of the wheel for the entire diagram consists simply of $\int \mathbf{rd}\theta$.

This reasoning is seen to be entirely general and independent of the character of the path **OAV,** and hence must be true whether it be the arc of a circle, a straight line or any other path.

In case the curve occupies only part of the revolution, as **ABC,** then it is clear that in going from **A** to **C** the record will involve the two parts, "1" and "2" above, and that the latter will remain included in the final result and will represent the summation of the record due to the elements of **OAV** between **A** and **C.** This obviously will be the value of $\int \mathbf{rd}\theta$ for the arc **GC** and it will be canceled by carrying the tracing point of the instrument back from **C** to **G.** This method of reasoning is **independent of the extent of the arc and is therefore equally true for an entire revolution, even when the diagram does not end at the same radial distance, as at the beginning.** In such cases it is necessary only to trace along the arc **OAV** so as to "close" the curve, thus canceling part "2" above and finding directly the value of $\int \mathbf{rd}\theta$ for a whole revolution.

In all cases the correction for part "2" of the record is made by tracing from the terminal point of the curve along the path, representing no change of time to a point lying in a circumference passing through the initial point. This may be stated in other words by saying that to eliminate part "2" of the record the tracing point must start and finish at the same distance from the center, and if the diagram is not of the kind to satisfy this condition then the necessary portion of a path of zero change of time must be used to supplement the diagram. This discussion is independent also of the nature of the curve **OAV.** It may be stated, however, that when **OAV** becomes a straight line the value of the correction becomes zero.

CHAPTER V

ENGINE INDICATORS AND REDUCING MOTIONS

The engine indicator is simply an instrument showing by graphic diagrams the variations of the pressure in the engine cylinder of steam, gas, air, or whatever the working substance may be. Before James Watt invented the engine indicator (about 1814) he had already used a steam pressure gage on the cylinder of his engine, and since the movement of the piston in the early steam engines was very slow, he was able to observe with his eyes how the pressure varied during a stroke of the piston. In modern engines the movement of the piston is so rapid, however, that a recording instrument is absolutely necessary.

Watt's indicator is illustrated in **Figs. 78** and **79**. It consists of a cylinder **CC** (**Fig.** 79) in which the piston **P** is moved against the resistance of the spring **S** by steam pressure from the engine cylinder; this pressure being exerted, of course, on the lower side of the piston. A pencil attached to the upper end of the piston rod traces on a sheet of paper a diagram **DD,** of which the height on any ordinate is proportional to the pressure. The paper is moved back and forth on a slide by a string **E** moved in conformity with the piston. The instrument was of great service to Watt in perfecting his steam engines. In the modern indicators, of which a few of the best known makes are to be described, there are many improvements over the instrument used by Watt.

Thompson Indicator. Of the engine indicators now in general use the Thompson is the oldest and best known. **Fig. 80** shows one view of this instrument and **Fig. 81** shows the corresponding sectional drawing.

It consists in essential parts of a piston, **8** (**Fig. 81**) moving in a cylinder **4.** This piston is rigidly connected to the rod **12,** which passes up through the cap **2.** The motion of the

piston rod 12 is transferred to the pencil 23 by means of suitable links designed to make the pencil move parallel to but usually four times as far as the piston **8.** The maximum pressure of the pencil on the paper used for the diagram is adjusted by the thread and set screw on the handle attached to the bracket **X.**

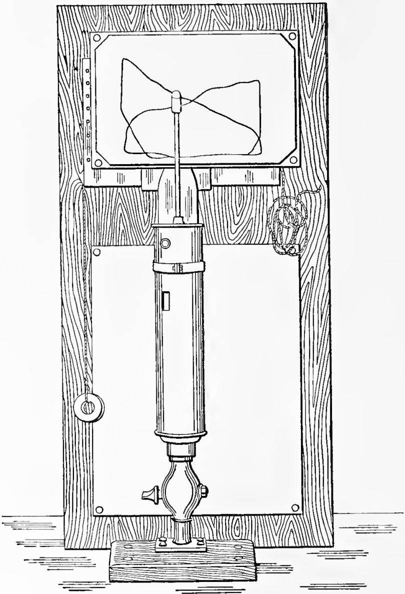


Fig. 78.—Watt's Original Steam Engine Indicator (Type of 1814).

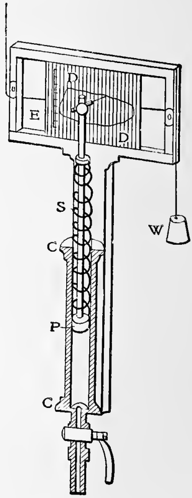


Fig. 79.—Section of Watt's Indicator.

The method of changing the springs in the various common forms of engine indicators should be well understood by everyone likely to be called on to "indicate" engines. When the work of changing springs is done clumsily or carelessly, a great deal of time is often wasted by the whole party engaged in the test. The method to be followed in changing springs of a Thompson indicator may be stated briefly as follows: The milled-edged

cap **2** should first be unscrewed from the top of the cylinder containing the spring and piston. This cap, together with the piston rod, piston, and link can then be lifted from the main body of the indicator. By unscrewing the small milled-headed screw **19**, connecting the piston rod with the pencil arm the spring can then be unscrewed, first from the cap **2** and finally from the piston **8**. By exactly reversing the operation another

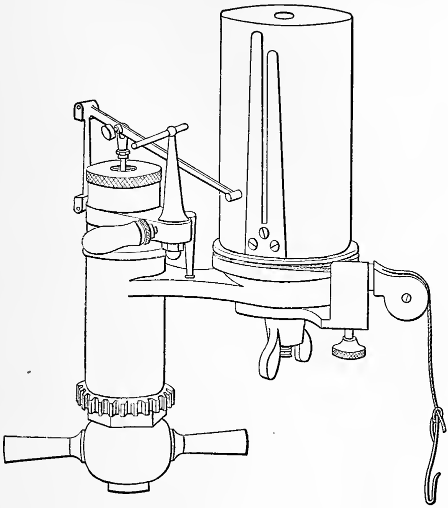


FIG. 80.—Thompson Indicator.

spring can be put in the place of the one removed. Changing springs in this instrument is a simple operation. No wrenches or other tools are required. Care should be taken, of course, to screw up the spring firmly against both the cap and the piston. Probably one-half the troubles with indicators in operation arise from loose springs, although not so often probably, with Thompson indicators as with some other types.

In selecting a spring for an indicator test it should be of such a scale that the largest diagram to be taken will not be more than $1\frac{3}{4}$ inches high; that is, if the maximum pressure will

be about 140 pounds, a spring with a scale of 80 pounds per square inch should be selected. The tension of the spring inside the drum carrying the paper for the diagram is varied by loosening the thumb nut and turning the large milled cap till the proper adjustment is secured.[1]

Crosby Indicators. For high-speed engines and for accurate results the Crosby indicator has long been a favorite with engineers. This indicator is illustrated in **Figs. 82** and **83**. It consists of a piston **8**, moving in the cylinder **4**, and is connected by means of the piston rod **10** and the link **14** to the pencil lever **16**. All of the pencil mechanism arranged to move the pencil point **23** in a straight line parallel to the motion of the piston **8**, is supported by the links **13** and **15** on the sleeve **3**. The indicator spring is fastened at its lower end to the piston by a ball-joint and at its upper end it is screwed into the cap **2**. The method of attachment of the springs to the piston by means of the ball-joint is shown more in detail in **Fig. 84**.

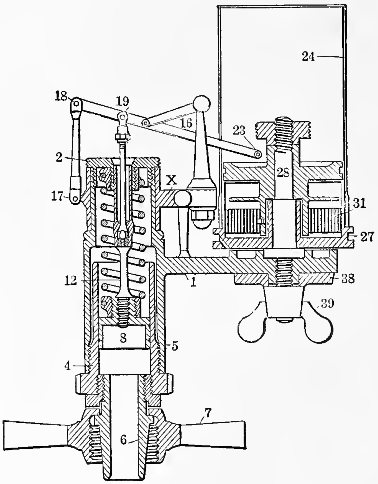


FIG. 81.—Section of Thompson Indicator.

In this indicator the spring is changed by first unscrewing the milled cap **2**, then this cap, the sleeve **3**, the piston rod **10**, and the connected parts can be removed from the cylinder **4**.

[1] Unless there is a very good reason for a change in the tension of the spring in the drum it should not be altered. Particularly in indicators which have been used a long time the pin holding the spring in place is likely to be much worn, so that if adjusted often the spring may get loose, and then there is often considerable difficulty in getting it again into its proper position.

By unscrewing the spring by hand from the cap, which, of course, must be prevented from turning, and also from the screw on the swivel head **12**, the piston, the spring and the hollow piston rod **10** are detached from the other parts. A socket-wrench of the special form provided in every indicator box of this make is to be slipped over the piston rod to engage with the small nut shown in **Fig. 83** at the lower end of the piston rod

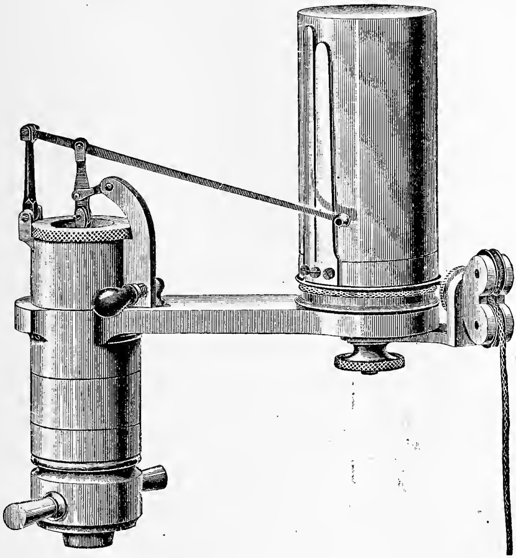


Fig. 82.—Typical Crosby Indicator.

10. Then the piston rod is readily unscrewed from the piston at the same time the spring in the indicator is released from its attachment to the piston. Now with the piston rod still in the socket of the wrench, slip the spring to be used over the piston rod until the head of the spring rests in the concave end of the rod. To do this, the wrench must be held upright, and then if the piston is inverted, or, in other words, if it is held so that the end screwing into the piston rod points downward the piston rod is ready to be screwed into the piston so that the trans-

verse wire of the spring passing through the bead will be held firmly in the slotted portion of the **socket** in the piston. Finally screw the piston rod firmly[1] into place. Before the last operation, the lower piston-screw (**Fig. 84**) should be loosed slightly, and afterward it should be screwed up lightly against the bead to prevent lost motion. It should not be screwed so tightly,

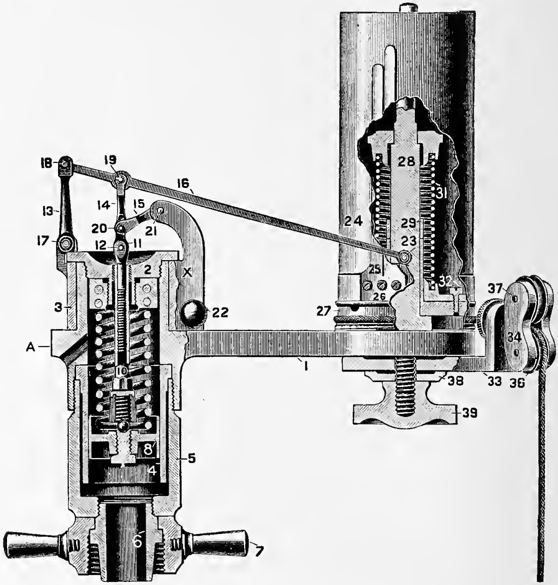


Fig. 83.—Section of Crosby Indicator.

however, as to prevent the bead from turning, otherwise the desirable qualities of the ball-joint for securing perfect alignment are lost. Now when the piston-rod spring and piston are again assembled, if the sleeve **3** and the pencil motion attached

[1] Special care should be taken when putting a spring into a Crosby indicator that the piston rod is screwed into its socket in the piston **P** (**Fig. 84**) as far as it will go; that is, until the extreme upper end of the socket **a a** is brought firmly against the bottom of the corresponding annular channel **b b** in the piston rod **R.**

to it are held in an upright position, the hollow piston rod can be slipped over the threaded portion of the swivel head **11** until the threads on the upper end of the spring engage with those on the cap **2.** Then the spring can be screwed securely into the cap **2.** Then permit the cap to turn in the sleeve **3,** and by still turning the spring, screw the piston rod on the swivel head **12,** until the top of the rod is nearly flush with the shoulder on the swivel head. The piston and attached spring are now ready to be put into the cylinder by slipping the sleeve **3** into position and screwing down firmly the cap **2.**[1]

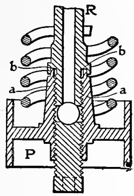


Fig. 84. — Section of Crosby Indicator Spring and Piston.

The height of the pencil cannot be adjusted to change the position of the atmospheric line without removing the piston from the cylinder of the indicator. It must be done, however, by unscrewing the cap **2** from the cylinder and removing it together with the sleeve **3** and the pencil mechanism. By turning the cap clockwise, the swivel-head **11,** and consequently also the atmospheric line is lowered. By turning in the opposite direction both are raised. Never try to make adjustments by removing or loosening the pins or screws at the joints, **17, 18, 19, 20** and **21.** These joints should always be kept tight enough to prevent any lost motion, and occasionally they should be lubricated with refined porpoise oil of the kind usually supplied with indicators.[2]

[1] Persons in charge of tests should always inspect indicators before the steam pressure is put on the springs to observe whether the cap has been screwed down firmly, and whether the pencil mechanism has been adjusted so as to give with a suitable spring a diagram of the proper height on the drum.

[2] Inexperienced testers often put the spring and piston into place by merely slipping on the sleeve **3** and without screwing down the cap **2.** Then, as a result, when the steam pressure is put on the indicator the piston, spring and pencil mechanism are thrown off with a great deal of force, and some of these parts are sometimes completely demolished.

When using an indicator having the spring inside the cylinder **4**—and this is true particularly in the Crosby Indicator—all adjustments should be made before the steam is turned on the indicator, because the piston, spring and cap soon become very hot, and unless the parts are cooled, preferably by dipping into cold water, they are difficult to handle.

The tension of the spring **in the drum** is changed very much more conveniently than in most other indicators. For high-speed engines the tension must be considerably greater than that required for those running at low speeds. The tension is adjusted by removing the drum (24) by a straight pull, and

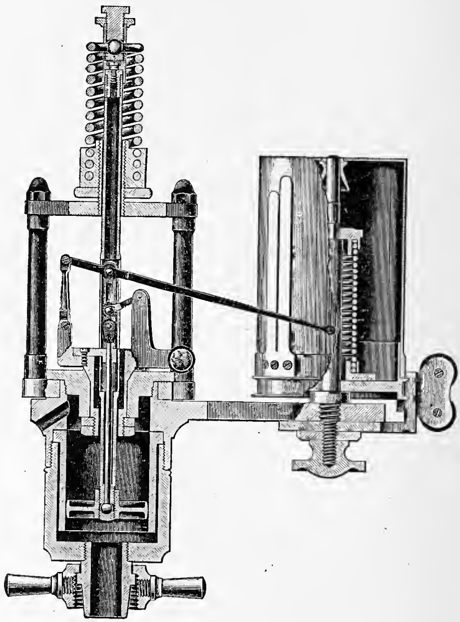


FIG. 85.—Crosby Outside-spring Indicator.

turning the knurled nut at the top of the spring (31) after lifting it from its square seat.

Crosby Outside-spring Indicator. Indicators with springs outside the cylinder (**Fig. 85**), so that they are not subjected to high temperatures, are particularly desirable for use with engines using superheated steam. There are two principal advantages: (1) The spring can be changed without removing the piston, avoiding an operation often causing confusion and loss of time; (2) the

tension of the spring cannot be affected by exposure to very high temperatures. The spring can be changed when the thumb-screw at the top of the central spindle has been unscrewed.

Star Brass Indicator—Navy Pattern. The indicator called the "Navy Pattern," manufactured by the Star Brass Co., is shown in **Fig. 86.** In general principles of construction it is like the Crosby indicator illustrated in **Fig. 85.** The most essential difference is in the type of straight-line parallel motion for the pencil lever. It will be observed that this is practically the same as that used in the Thompson indicator (**Fig. 80**).

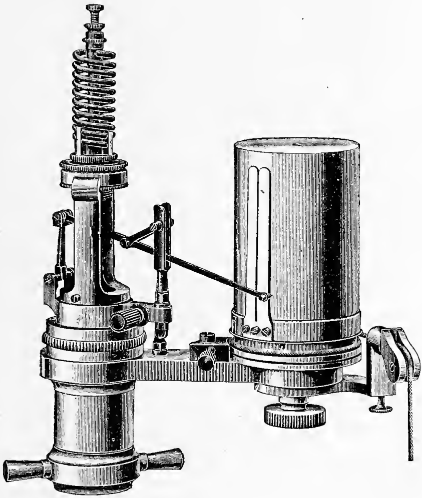


Fig. 86.—Star Brass Indicator—Navy Pattern.

Tabor Indicator. In the form in which it is now manufactured the Tabor indicator, **Fig. 87,** differs from indicators like the Crosby particularly in the means employed for producing a straight-line parallel motion for the pencil. This is accomplished by the movement of a roller attached to the pencil lever in curved slots on the inside of the rectangular box-shaped part shown in the figure, attached to the cylinder cap.

As regards the point of flexibility in the mechanism, this is not between the spring and the piston, but, more like the Thompson, is between the piston and the piston rod. Details of this construction are shown in **Fig. 88.**

To Change the Spring. The cylinder cap must be first unscrewed, and then this cap, together with the piston, spring, and connected parts can be lifted from the cylinder of the

indicator. By removing the small screw under the piston the latter can be unscrewed from the lower end of the spring. The other end of the spring can then be unscrewed from the cylinder cap. Another spring is put into the indicator by slipping it over the piston rod with the end stamped **T** upper-

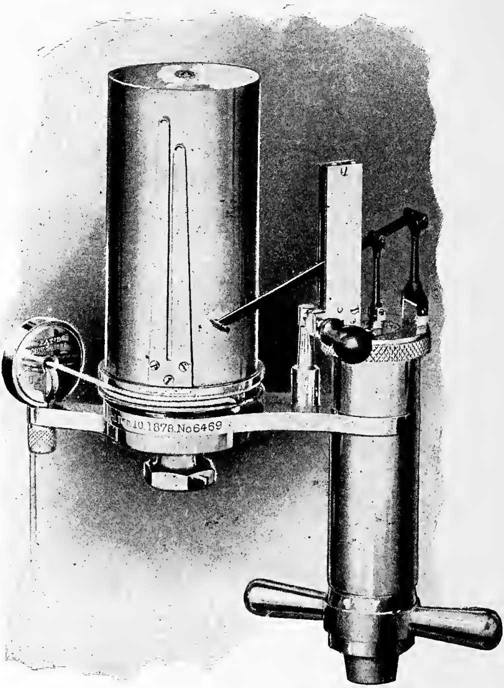


FIG. 87.—Tabor Indicator.

most, screwing this end into the cylinder cap and screwing the piston to the lower end. The pencil mechanism must be moved downward until the piston rod enters the piston and the square shoulder enters the corresponding square socket in the piston. In this last operation care must be taken that

the rod is firmly and accurately in the hole, and then the screw at the bottom of the piston should be firmly applied.

To Change the Tension of the Spring in the Drum. The drum itself is first removed. Then after loosening the knurled nut on the central shaft and after the drum carriage has been lifted clear of the stops, the carriage can be turned in the

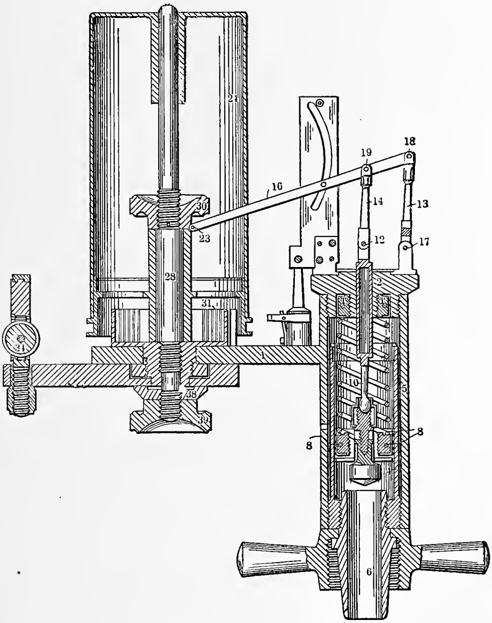


Fig. 88.—Section of a Tabor Indicator.

required direction to secure the necessary tension and it can then be replaced by lowering into the stops. Care must be taken also that a firm grasp on the drum carriage is not lost, otherwise the spring will become uncoiled and probably also detached.

Bachelder Indicator. Fig. 89 illustrates an engine indicator which is in many essential parts entirely different from the

general type to which all those already described belong. It is so simple in construction that scarcely any description is necessary. The most radical difference is, however, in the form of spring used. This is flat and is arranged with a movable fulcrum which can be adjusted to change the scale of the spring. Although a wide range is obtainable in this way, it has been found unsatisfactory to attempt to use the single spring for all the ranges from the highest pressures to low vacuums. On

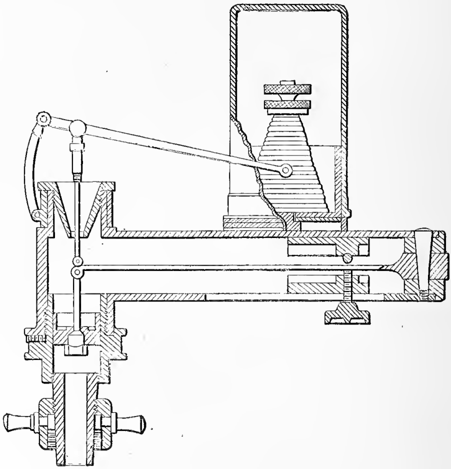


FIG. 89.—Bachelder Indicator.

this account at least two springs, one for high and the other for low pressures, are usually supplied.

Springs are changed by first removing the taper screw shown at the extreme right-hand side in the figure, and then after unscrewing a circular cap on the side of the cylinder the pin connecting the spring to the piston rod can be withdrawn with a small pliers or similar instrument. Usually before the spring can be withdrawn the thumbscrew attached to the movable fulcrum must be loosened. In the ordinary operation of the instrument the piston is not removed.[1]

[1] When the spring is calibrated, the piston should be taken out so

The spring on the drum is conical in form and is adjusted in practically the same way as in the Crosby Indicator.

Precautions for Care of an Indicator. Unless an engine indicator is well taken care of, very soon it will be in a condition in which no reliance can be placed on results obtained with it. That the necessary precautions should be taken is all the more important, because it is one of the most expensive as well as the most delicate instruments used by an engineer in his ordinary practice. The following precautions are particularly important:

1. Before an indicator is used all the working parts, especially the piston, should be carefully cleaned. Then after a spring suitable for the pressure has been attached in its proper position and a little cylinder oil has been smeared in a thin coat on the working surface of the piston the parts should be replaced. Moving parts of the pencil mechanism should be oiled occasionally with watchmaker's or porpoise oil.

2. Adjust the screw on the handle provided for moving the pencil so that when the pencil is sharp the application of the usual pressure on the handle will give a very fine line.

3. Adjust the length of the indicator cord so that the drum will be neither too loose nor too tight; or in other words, so that the drum will not strike either of the stops when the engine is operating. On a small engine this is most easily tested by observing the card when the engine is on each of the dead-centers. If the card is either too long or too short the drum will not be moved in either case the required distance, and the indicator card will be correspondingly too short, therefore inaccurate. The cord used should be selected with care. It must be of such a quality as not to be stretched appreciably by the forces to which it is subjected.

4. The atmospheric line should always be taken, preferably after the diagram has been made. It is drawn when the indicator cock is closed. By this order of procedure in tests, the diagram can be more easily taken exactly "on the signal." For calculations the length of the diagram must always be measured on the atmospheric line or on a line parallel to it. The indicator

that a little cylinder oil can be put on it. It is not so necessary when in use on a steam engine, as the oil in the steam will usually provide sufficient lubrication.

cock should be kept closed and the cord to the reducing motion should be unhooked except when a diagram is to be taken. When the cord is unhooked the drum should not be permitted to snap back against the stop.

5. **Immediately after a diagram has been taken it should be removed from the drum and examined.** If there are unusual irregularities in the lines, unaccountable differences in the areas or in the lengths of different cards, the facts should be noted and the best efforts should be made to remedy the faults. Irregularities are usually due to stretching of the indicator cord, grit on the piston, lost motion in the working parts, usually inside the indicator cylinder, or excessive friction caused by overheating of the piston, particularly when used on gas engines.

To correct these faults concerning the piston it must be removed from the cylinder and should then be carefully cleaned and lubricated again with cylinder oil. Before putting the piston and connected parts back into the indicator cylinder it should be observed whether or not all the parts are connected firmly and without lost motion.[1]

6. After a test, the indicator should be removed immediately from the engine, protecting the hands with waste or thick gloves to prevent burns. All the parts, especially those in the cylinder, should be thoroughly cleaned and then put together again without the spring, which should be put away with the other springs in a box provided for the indicator. An indicator should never be handled by taking hold of the drum, as usually it is fastened to the indicator by only a loose slipjoint, and this comes off easily.

[1] One of the causes of errors in results obtained with indicators not so readily detected is due to the pencil motion not being parallel to that of the piston in the indicator. A simple test for this is to draw an atmospheric line on a card placed on the drum. The card should be at least as wide as the height of the drum. Then after taking out the spring raise the pencil to the full height of the card by pressing lightly on the piston. This operation should be repeated several times at several points along the length of the card. To secure the best accuracy it is desirable to "block" the drum in each position. If the lines drawn are exactly perpendicular to the atmospheric lines there is no error in the pencil mechanism. If the test for perpendicularity is made by a triangle and straightedge, it should be done with the triangle first lying on one side and then on the other, to eliminate any inaccuracy in it. Often the triangles used by engineers are very inaccurate.

SPECIAL TYPES OF ENGINE INDICATORS

Cooley-Hill Continuous Indicator. For many purposes of investigation it is very important to have continuous records showing the variations of the cycles in the operation of an

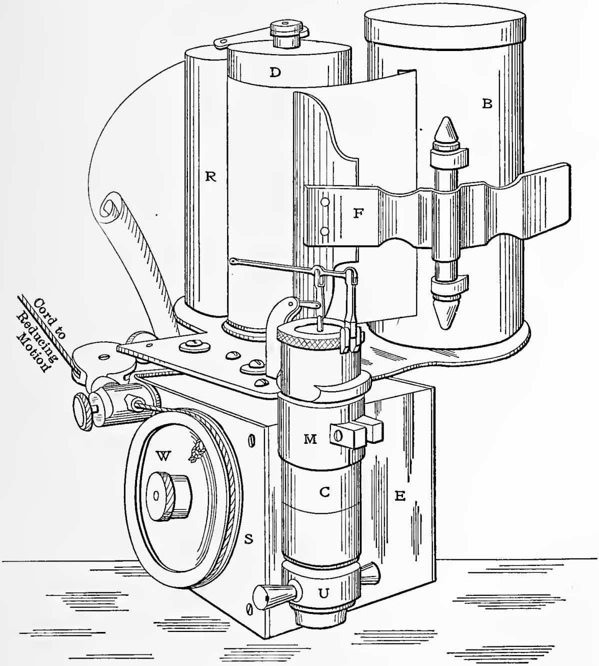


FIG. 90.—Cooley-Hill Continuous Indicator.

engine. Many devices have been used for this purpose, but as the motion was taken from the crank shaft there was no simple relation between points on these diagrams and the corresponding points in the stroke of the engine. Furthermore, because of the difficult relation, such cards could not be measured with a planimeter. Similar apparatus for the same purpose operated by an electric motor were open to the same objection. To

overcome these difficulties a continuous indicator was developed in the Engineering Department of the University of Michigan in which the motion was proportional at every instant to the movement of the piston. With a diagram obtained with this instrument it is not difficult to determine the dead-center following release, and the conventional indicator card for an engine is then readily obtained by turning the diagram for the complete cycle back on itself by folding the card or ribbon at this dead-center. If transparent paper is used, the complete diagram can be seen with all the points in their true relative positions as regards the movement of the piston. The indicated horse power can then be readily calculated with the aid of a planimeter. This continuous indicator is illustrated in **Fig. 90.** The indicator cylinder **C,** the piston, and the pencil motion may be of any standard make, as the **collar M,** for attaching the drum mechanism, is adjustable in size so that it can be fitted to indicator cylinders of different diameters. By this arrangement only one drum motion need be provided for using this indicator motion on a number of types of indicators such as would be required for use with steam engines, gas engines, high-pressure air compressors, ammonia compressors, etc. In this apparatus the drum **D** moves forward a given amount with every stroke of the engine. The indicator cord **S** is connected to the indicator reducing motion, and is driven by being connected in the usual way to the cross-head of the engine.

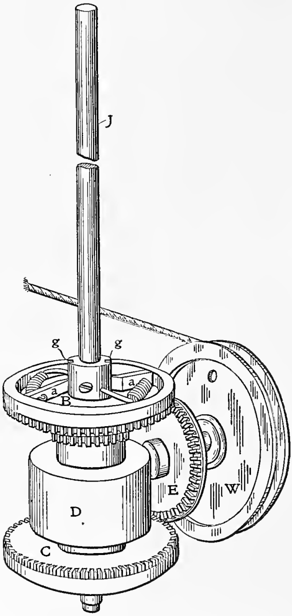


Fig. 91.—Details of Cooley-Hill Indicator.

The mechanism operating the drum motion is illustrated in **Fig. 91.** It consists essentially of two miter wheels **B** and **C,**

meshing with a similar wheel **E,** to which the pulley **W,** carrying the indicator cord, is attached. At the top of the wheel **B** and at the bottom of **C** are so-called silent ratchet clutches, **a, a,** each of which operates in only one direction to grip the collars concentric with the wheels **B** and **C.** Only one of these collars is shown in the figure. Both are rigidly attached to the central spindle **J,** carrying the indicator drum **D** (**Fig. 90**). For example this central spindle is gripped by the ratchets **a, a** in the wheel **B,** during the "forward" stroke of the engine, and is released during the "back" stroke. The ratchet in the wheel **C,** on the other hand, grips this spindle during the "back" stroke and releases on the "forward" stroke. In this way the drum **D** is constantly moved on the spindle **J** in the same direction. Neither of the wheels **B** nor **C** is directly attached to the central spindle, and they can move it only when they move in the direction in which they grip their ratchets **a, a,** engaging in the grooves **g, g.**

The miter wheels **B** and **C** are connected to each other by means of a spiral spring enclosed in the casing **D.** This serves the function of the ordinary drum spring in the usual type of indicator for bringing the drum and string back when the cross-head moves toward the indicator.

Optical Indicators. The usual types of indicators operating with a piston are not suitable for engines running at much over 400 revolutions per minute. For higher speeds optical indicators are used. These operate by the deflection of a beam of light from a mirror, the deflection being proportional at any instant to the pressure. When such a device is used on an engine successive indicator diagrams can be readily observed and compared by marking with a pencil the reflection upon a ground-glass plate, and if a photographic sensitive plate is exposed to the beam of light in the place of the ground glass, a permanent impression can be taken, showing at any instant the operation of the engine. Optical indicators are practically the only kind that can be used successfully for indicating the action of modern high-speed **automobile engines.** Every well-equipped automobile testing plant should be provided with one of these instruments. One of the simplest and best apparatus of this kind is illustrated in **Fig. 92.** The indicator is shown in the picture vertically above and connected to the head of the engine.

Steam pressure is communicated to the instrument through the usual type of indicator cock supporting it. A system of levers shown (a simple reducing motion) serves for reducing the length of the stroke of the engine to a suitable size for such a small instrument. A glass mirror moved about a vertical axis by the motion transmitted from the cross-head and about a horizontal axis by the pressure in the engine cylinder, reflects a

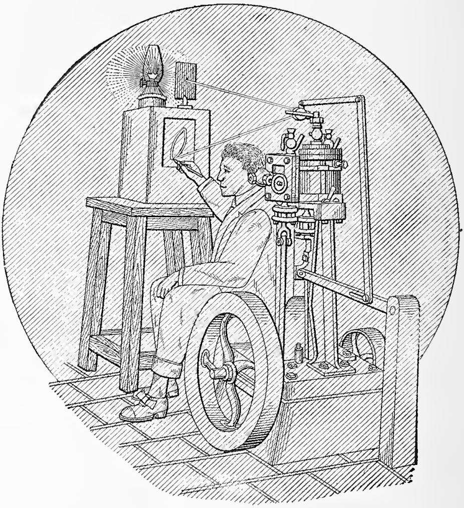


FIG. 92.—Perry's Optical Indicator.

beam of light from a lamp upon a sheet of paper so that the indicator diagram can be traced.

Details of the essential parts of this instrument are shown in **Fig. 93**. Through the indicator cock the pressure in the engine cylinder is communicated to the cored passages marked **A, A.** This pressure tilts the mirror **B,** attached to the thin steel diaphragm **D.** When, therefore, the mirror is still, a ray of reflected light will be seen as a bright **spot** on the screen; but when moved both by the pressure and the motion of the cross-head the conventional indicator diagram is traced. It is very

interesting to watch the rapid change of shape of such diagrams as load, speed, pressure, cut-off, etc., are changed. With such an instrument these interesting phenomena in engine operation can be illustrated on a ceiling to a large class of students.

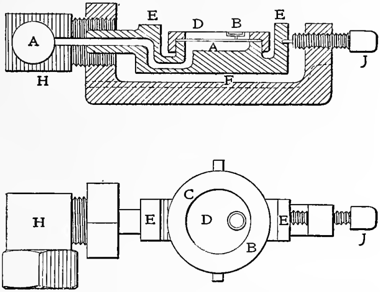


FIG. 93.—Essential Parts of Perry's Optical Indicator.

Another type of optical indicator intended particularly for high-speed automobile engines is shown in **Fig. 94.** In this instrument the movement of the beam of light is produced by reflection from a small mirror **M** arranged to move in two distinct planes at right angles to each other. In one plane

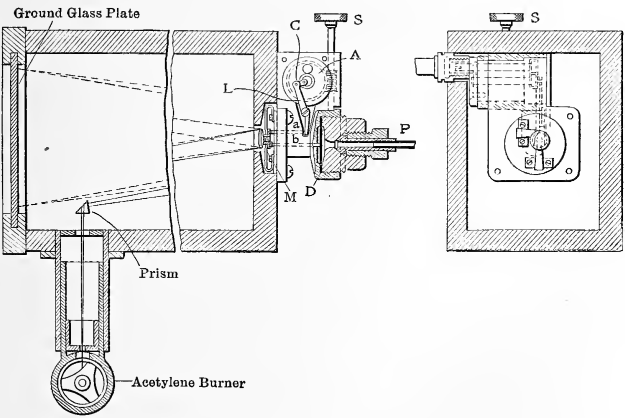


FIG. 94.—Section of a "Monograph" Optical Indicator.

the movement of the piston is accurately reproduced, and in the other the movement is proportional to the pressure. Either

of these movements or deflections of the mirror, taken alone, would cause the reflected beam of light to trace on the ground-glass plate a straight line; that due to the pressure being arranged to produce a straight vertical line and that due to the motion of the piston a straight horizontal line. But obviously the two movements taken together trace a diagram indicating at any instant the pressure in the engine cylinder for the corresponding position of the piston. A flexible shaft, attached at one end to the crank shaft of the engine, moves the disk **A** and with it the crank **C**, as well as the small lever **L** attached to it. The free end of this lever is arranged to turn the mirror **M** about a vertical axis by means of the small strut **a**, while the pressure exerted on the diaphragm **D**, as transmitted from the engine cylinder by the pipe **P**, moves the mirror about a horizontal axis by means of the strut **b**. In this apparatus the diaphragm takes the place of the piston and spring in the ordinary type of indicator. These diaphragms, like those used in pressure gages (see page 8) can be made of such thicknesses that a diagram of satisfactory size can be obtained for high or low pressures. When the diaphragms are carefully calibrated, a reasonable degree of accuracy can be expected. The relative motions of the mirror in the two planes are set in phase by adjusting the milled screw **S**, operating a small worm wheel serving for changing the angular position of the crank disk **A**, to make the movement of the mirror about the vertical axis correspond with that due to the pressure. **Fig. 95** shows the apparatus as it would be set up for indicating an engine. A sample indicator card taken with this apparatus from a gasoline automobile engine is shown in **Fig. 96**.

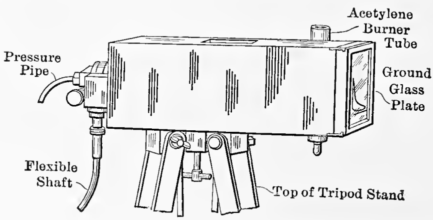


Fig. 95.—"Monograph" Optical Indicator Ready for Attachment to Engine.

Calibration of Indicator Springs. The pistons of engine indicators are invariably made of a very definite area, **usually**

one-half square inch; and it is possible to calibrate the deflection of the springs with respect to this area, so that a certain definite pressure per square inch [1] in the cylinder will correspond to a definite deformation of the springs. In English units the pressure on the piston in pounds per square inch corresponding to a movement of the indicator pencil on the diagram of one inch is called the **scale**[2] **of the spring.** Indicator springs should always be calibrated by the makers. The calibration should be made when they are in the indicator in which they are to be used.

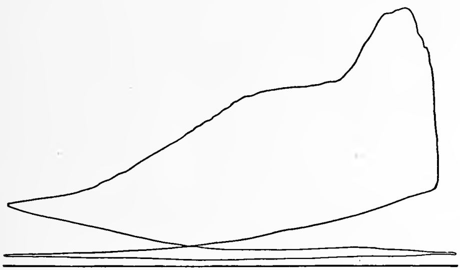


Fig. 96.—Indicator Card taken from an Automobile Engine with Optical Indicator.

Cooley Apparatus for the Calibration of Indicator Springs. An apparatus similar to the one designed by Professor M. E. Cooley of the University of Michigan is most generally used for the calibration of indicator springs. One of the latest and more elaborate forms of this instrument is shown in **Fig. 97.** In its essential parts this apparatus consists of a small cylinder **C,** supported on a bracket **B,** a connection **I,** at the top of this cylinder for the attachment of the indicator to be tested, and a stuffing-box or gland at the bottom of the cylinder into which a plunger-piston **P** is fitted. The lower end of this plunger rests on a sensitive platform scales. Any pressure in the cylinder **C** can therefore be weighed. Steam is admitted to the cylinder through

[1] Indicators are always designed to relieve the pressure above the piston due to leakage around it, so that on this side there is always atmospheric pressure.

[2] Instead of "scale" the word "number" is often used. That is, a spring of which the scale is 40 pounds would be called "No. 40."

a pipe **E,** and is exhausted through the pipe **A.** By adjusting the globe valves on the pipes **A** and **E** any pressure desired can be secured in the cylinder **C,** and this same pressure is, of course, exerted both on the piston of the indicator above and on the plunger **P** below. This plunger is usually made with an area of one-half square inch. For a plunger of this area, then, if for a given pressure the scales balance at 10 pounds, the pressure in the cylinder **C,** and on the piston of the indicator, is 20 pounds

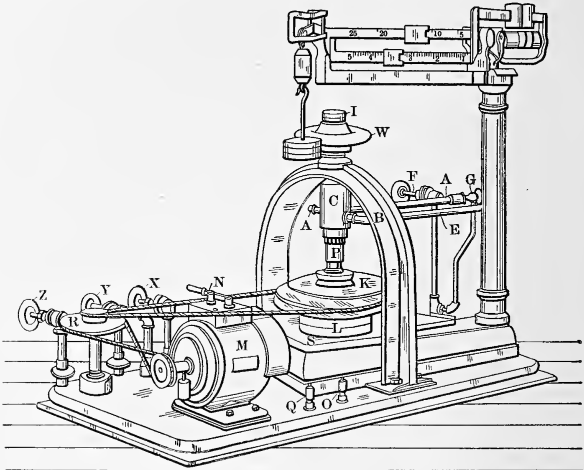


Fig. 97.—Apparatus for Calibrating Indicator Springs.

per square inch. To eliminate friction as much as possible, the plunger **P** should be kept spinning when observations are being taken. For this purpose a hand wheel **K** with considerable mass, for its "fly-wheel" effect, is provided on the shaft of the plunger. A more uniform motion of the plunger is obtained, however, by having the hand wheel grooved to take a small belt to be driven by an electric motor **M.** The plunger is supported usually on a ball-bearing joint set in a low pedestal **L.**

By connecting a pipe **E** to a suitable manifold or similar fitting, to which are attached three separate pipes supplying respectively steam, air, and water under pressure, an indicator

can be tested with varying pressures under the actual conditions in service; that is, when used for steam, air or water.

A simpler form of the Cooley apparatus intended for the so-called "dry method" of testing is shown in **Fig. 99.** A

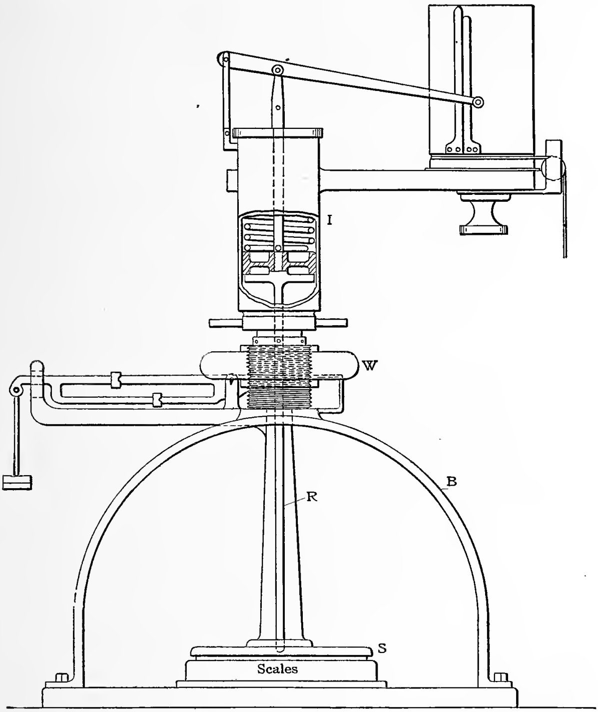


FIG. 99.—Apparatus for "Dry" Method of Indicator Testing.

suitable fitting for receiving the indicator **I** is supported on the bracket **B.** The legs of this bracket span over a sensitive platform scales **S.** A small rod **R** rests at its lower end on a small pedestal standing on the platform of the scales **S.** On the top of this rod there is a cap supported on a small conical bearing to

give some flexibility. This cap is made to fit easily into the lower side of the piston in the indicator. The indicator itself is attached to the top of the hand wheel **W.** Then when the hand wheel is screwed downward the indicator comes down with it and compresses the indicator spring. At the same time a pressure is exerted on the rod **R** which can be balanced on the scale beam. When a force is applied to compress the spring in the indicator, the magnitude of the force can be determined by weighing the pressure on the scales. If the area of the piston in the indicator is one-half square inch, then twice the weight on the scales is the pressure exerted in pounds per square inch. Heat can be applied to the indicator by passing steam through a rubber tube wrapped around the cylinder.

Method for Calibration of Springs. After cleaning the internal parts of the indicator, inserting the spring to be calibrated, and oiling the piston with cylinder oil, the indicator is to be attached to the indicator cock on the calibrating apparatus. Before putting the card on the indicator drum on which the record is to be made, two approximately parallel and vertical lines should be drawn on it about one-half inch apart, similar to the lines **AB** and **CD** in **Fig. 100.** Meanwhile the indicator should be thoroughly warmed if a calibration with steam pressure is to be made. Then with the indicator cock and the valve on the steam pipe **E** closed and the one on the exhaust pipe **A** open, draw the first calibration line on the card. This should be made by setting the pencil point at **D,** and then by pulling the cord attached to the drum draw a line **crossing** the vertical line **AB.** With springs of which the scale is 30 pounds or less, a similar record should be made for increments of every 5 pounds per square inch change in pressure, while for higher scales the increments may be made 10 pounds. If with increasing pressures the lines are drawn toward the left, then with decreasing pressures they should be drawn toward the right with equal increments, beginning at the opposite vertical line **AB.** By this method the corresponding lines for equal pressures will be immediately over each other between the two verticals. With an accurate scale, graduated preferably to one one-hundredths inch, measure between **AB** and **CD** the distance from the atmospheric line first drawn to the various "pressure lines," and record the results. Care should be taken

that with the increasing increments the pencil **rises** to the required pressure and that with decreasing increments it **falls** to these pressures. In other words, if when the lines for increasing pressures are being drawn, the pressure rises too rapidly to draw the line at the proper time when the scale beam is just balancing, then the pressure should be again reduced below the value required, so that the pencil will be again ascending when the line is drawn. Similarly for decreasing pressures, if the pressure gets too low, it must be increased and again brought down to the required value.[1] If the results

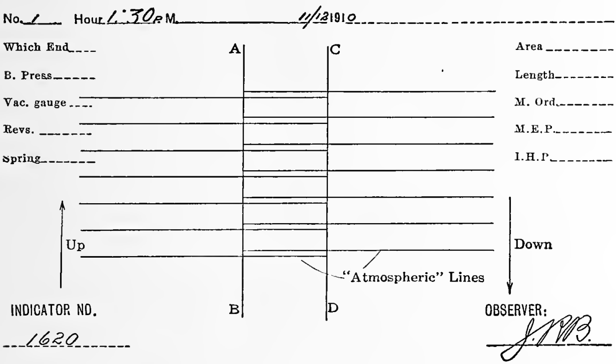


FIG. 100.—Sample Card Illustrating a Test of an Indicator Spring.

obtained do not seem to be consistent, the difficulty is probably due to passing the required pressure so rapidly that the lines have not all been drawn at the proper time.

The difference between the lines for increasing and decreasing pressures shows the amount of friction and lost motion in the indicator.[2] The error of the instrument is obtained by comparing the mean ordinates of the card thus obtained with the actual

[1] The same precaution must be observed in beginning the test. To be sure that the pencil and piston have not been falling instead of rising, the piston rod should be pushed down lightly before the atmospheric line is drawn.

[2] Half of this difference, to be more accurate, represents the friction and lost motion in that position.

pressures as determined by weighing. From time to time the accuracy of the platform scales should be determined by testing with standard weights. For dependable results two calibrations, each "up and down," should be made for each spring and the results compared.

When indicators are used for pressures which are never less than atmospheric, the springs are in compression and apparatus of the form described are satisfactory; but when indicators are used on the low-pressure cylinders of engines the springs are usually in **tension.** For this service a slightly different device must be used for calibration. A suitable apparatus is shown in **Fig. 101.**

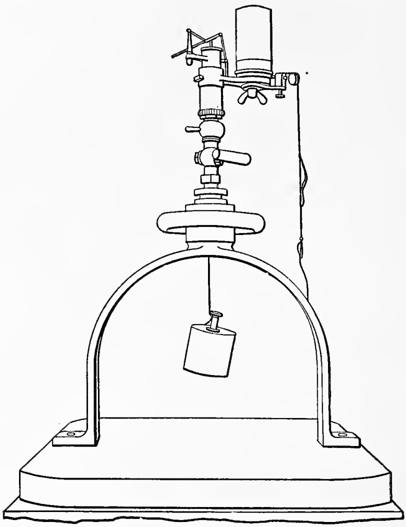


FIG. 101.—Apparatus for Testing Indicator Springs in Tension.

The indicator **I** is supported on a bracket similar to the one used in the apparatus shown in **Fig. 99.** A short steel rod (about No. 18, B. & S. gage) is attached to the lower side of the piston in the indicator by screwing into a hole tapped centrally. Now if weights are suspended from the end of this rod[1] the spring can be calibrated in tension by drawing lines on a paper

[1] The weight of this rod and wires or strings supporting the weights must be added to them to get the correct tension. If, however, only the true scale of the spring is desired, as is usually the case, the weight of these parts need not be considered, provided, of course, the atmospheric as well as the other lines are all drawn with these parts attached to the piston.

card placed on the drum in the same general way as when the spring was tested in compression.

A suggested form for the arrangement of data for these calibrations is given below.

CALIBRATION OF INDICATOR SPRING (COMPRESSION)

In Indicator No..............

Rated scale of spring..
Diameter of piston of testing apparatus...............................ins.
Area of piston of testing apparatus...............................sq.ins.
Identification marks on spring...

Observers { .... Date................
....

No. of Reading.	Weight on Scales, Lbs.	Actual Pressure on Piston, lbs./sq.in.	Ordinates or "Heights" Measured on Card, Inches.			True Scale of Spring, (3) ÷ (6).	Remarks.
			Up.	Down.	Average.		
1	2	3	4	5	6		

Curves. Results should be shown graphically for calibrations of indicator springs by plotting for abscissas the average height, inches, and for ordinates the corresponding actual pressures in pounds per square inch.

Calibration of Indicator Springs with the Mercury Column. The method to be followed in calibrating indicator springs with a mercury column is essentially the same as described on pages 18–22 for the calibration of pressure gages. After the indicator has been cleaned and oiled, it should be attached to the testing drum or cylinder by means of an indicator cock. Then while the indicator is being heated to the temperature of the fluid medium used (steam, air, or water), the paper card can be put on the drum after first drawing two vertical lines one-half inch apart, as explained when describing the Cooley apparatus. Following these same instructions the atmospheric and other pressure lines are drawn first with increasing and then with decreasing increments.

Testing the Drum Motion of Indicators. An apparatus for determining the relative accuracy of the drum motion of indicators as regards uniform tension in the cord for a given speed is illustrated in **Fig. 102.** This device, known as Brown's,

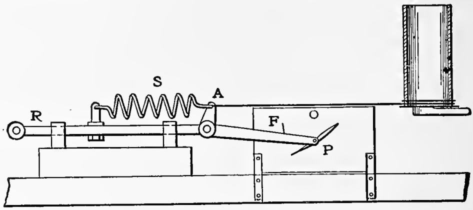


FIG. 102.—Brown's Apparatus for Testing Drum Motion.

consists of a rod **R,** which is made to take the same movement as the end of the cord in a reducing motion by being attached through a connecting rod to a crank pin on a disk like the face plate of a lathe. At the other end this reciprocating rod is attached to a bell-crank lever **F,** of which the outer end **P** carries a pencil for making a diagram on a card attached to a vertical frame **O.**

The short end **A** of the bell-crank is connected to a helical spring **S** and the other end of this spring is attached to an arm fitted on the rod **R.** To the end of the spring at **A** the indicator

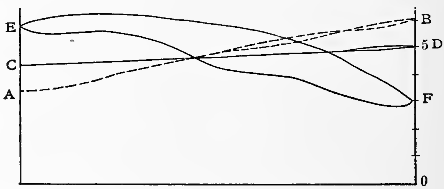


FIG. 103.—Diagrams taken from Apparatus for Testing Drum Motion.

cord is attached, being of the same length as when in use on an engine. Now when the reciprocating rod **R** is in motion and the tension in the spring **S** is uniform the pencil at **P** will describe a horizontal line. If, however, the tension in the indicator cord varies and consequently also the tension in the spring is not uniform, the pencil will describe a closed curve. Examples

of such curves are shown in **Fig. 103.** Curve **AB** was obtained when the apparatus was moving very slowly, **EF** when operating at about 700 revolutions per minute, and **CD** when the speed was about 250 revolutions per minute. The latter speed is obviously the one for which the stiffness and length of the spring in the indicator drum are most suitable.

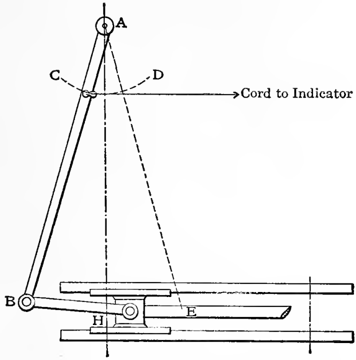


FIG. 104.—Simple Pendulum Reducing Motion.

Reducing Motions for Indicators. In the case of most engines the length of the stroke is very much longer than the greatest possible movement of the drum of the indicator. It is therefore necessary to provide some means called a reducing motion, which produces shorter movement, but which at every

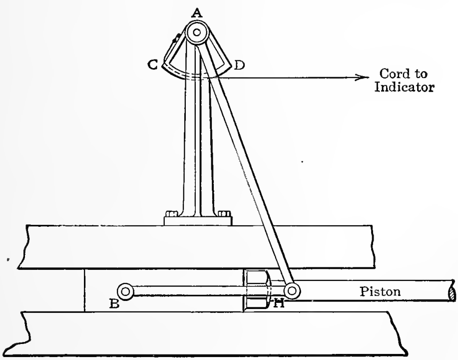


FIG. 105.—Pendulum and Quadrant Reducing Motion.

instant corresponds exactly with that of the cross-head. If this correspondence is not secured the length of the indicator

diagram cannot be accurately reduced nor calculated, and the timing of the events or so-called "points in the stroke" will not be correctly represented.

One of the simplest forms of reducing motions is illustrated in **Fig. 104**. This device is pivoted at one point **A** to a pedestal

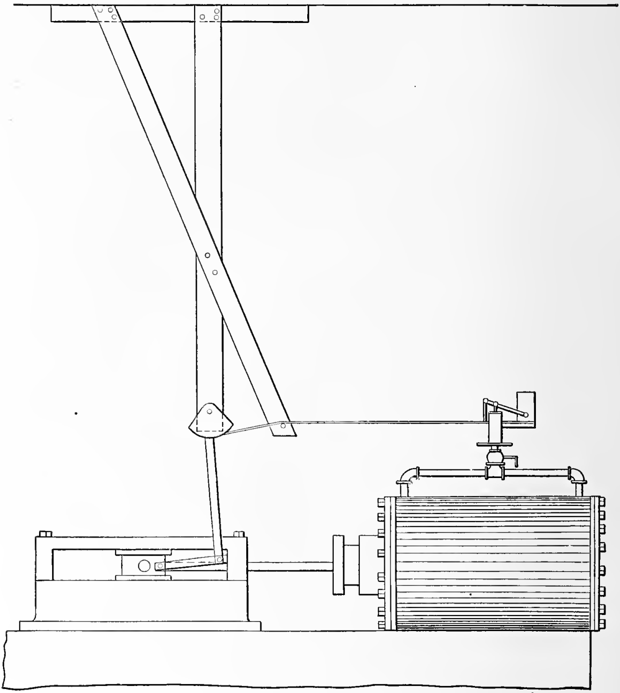


Fig. 106.—Brumbo's Pulley.

supported on the frame of the engine, and has a link **BH** connected to the cross-head. The indicator cord rides in a circular arc **CD,** proportioned to give the required movement to the drum of the indicator. Although this arrangement does not give an exact reproduction of the movement of the cross-head, yet if the pendulum **AB** and the cross-head are simultaneously

at the middle of their strokes the error is insignificant. An improved type of this device is shown in **Fig. 105,** in which the cord rides in a groove on the circumference of a quadrant pulley. By attaching the pendulum to the quadrant pulley by means of a suitably designed "slip" joint, the pendulum can be disconnected from the quadrant so that the indicator cord will be moved only when the indicator diagrams are to be taken.

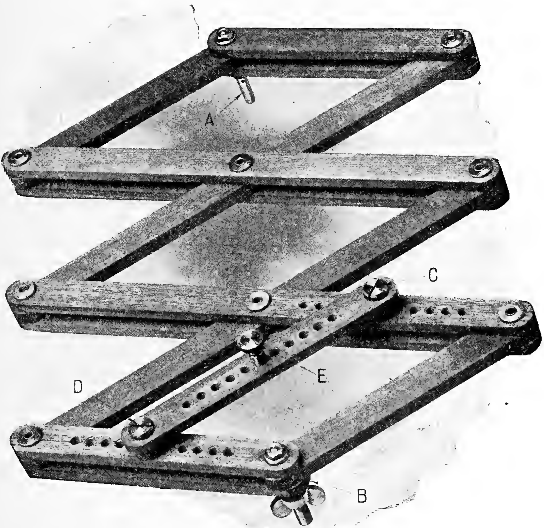


FIG. 107.—Pantograph or Lazy-tongs Reducing Motion.

Brumbo's Pulley is also a form of reducing motion of the pendulum type. It is illustrated in **Fig. 106.** In this device a guide pulley is placed between the indicator and the quadrant pulley. A modified and simpler device of the same kind consists of an upper portion moving in a vertical direction in a swinging tube and a lower portion pivoted directly to the cross-head.

Of the **portable** devices used for reducing motions the **pantograph, Fig. 107,** is probably the one most used. This device

is sometimes known as a **lazy-tongs.** Because of the numerous parts of which it is composed, requiring a great number of joints, it is likely to be troublesome with high-speed engines. A plan view showing one of the methods of attachment of this device to a horizontal engine is given in **Fig. 108.** This instrument when firmly put together is a **perfect** reducing motion.

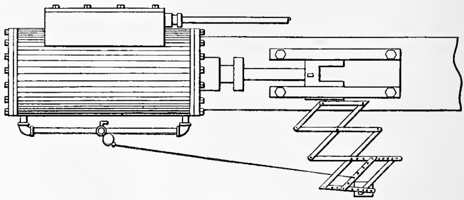


FIG. 108.—Plan View, Showing Attachment of Pantograph.

Parallel Motions, like the one illustrated in **Fig. 109,** are also very commonly used. They are made usually of rods of iron or of steel nicely riveted together at the joints. The indicator cord is generally attached at **B** and the ends **A** and **C** of the long rod are

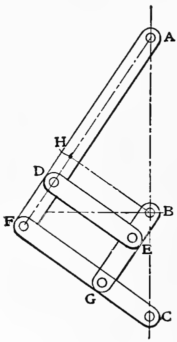


FIG. 109.—Simple Parallel Reducing Motion.

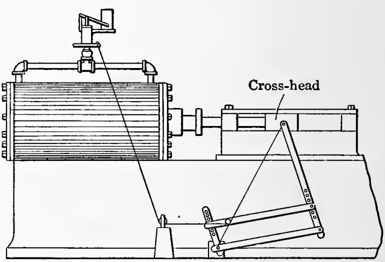


FIG. 110.—Simple Parallel Reducing Motion as Attached to a Steam Engine.

fastened respectively to the cross-head and to the frame of the engine. It is a necessary requisite that the points **A, B,** and **C** shall lie in a straight line as shown. Also **DE** must be equal in length and parallel to **FG.** Then **AF** is in the same ratio

to **HF** as the stroke of the piston is to the length of the indicator diagram.

Methods of attachment of similar devices to engines are shown in **Figs. 110** and **111.**

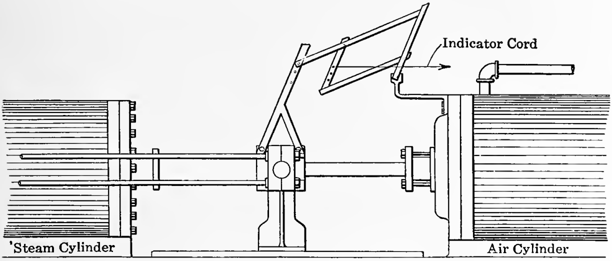


FIG. 111.—Simple Parallel Reducing Motion as Attached to an Air Compressor.

Fig. 112 illustrates another interesting parallel motion. It consists of a rod **R,** moving in a slide **S,** parallel to the piston-rod. A link **BD** is attached to the slide **R** at **B** and to **CE** at **D,** while **AE** is fastened at one end to the cross-head **C.** In this case again if **A, B,** and **C** are in the same straight line, then the following relation holds: **AE : BD** and **CE : CD** as the stroke of the piston is to the length of indicator diagram. The cord is hooked on a pin at **H.** It is desirable to have a separate pin for each indicator used.

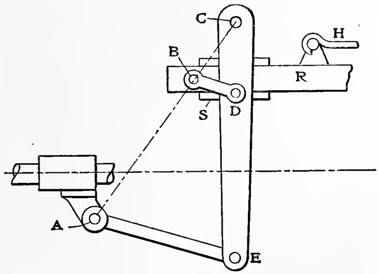


FIG. 112.—Sliding Type of Parallel Reducing Motion.

Reducing Wheels which consist simply of a large and a small pulley attached to the same axis, are coming into more or less general use. A typical arrangement is illustrated in **Fig. 113.** Pulleys **D** and **D′** are usually connected by a sliding sleeve so

that they can be disconnected when indicator diagrams are not being taken.

Fig. 114 is a device by Armand Stevart **for engines with long strokes. A** and **B** are fixed ends of cord wrapped around a

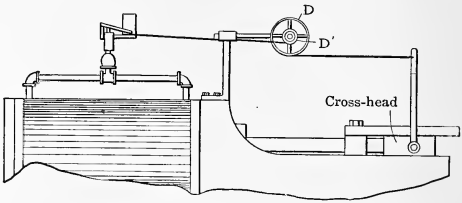


FIG. 113.—Reducing Motion of Concentric Pulleys.

pulley **D.** The indicator cord **g** is attached to a small pulley **D′** and passes around a guide pulley **G. D** and **D′** are attached to the cross-head **C.** Then diameter **D** ÷ diameter **D′** = stroke of piston ÷ by the difference between stroke of piston and length of card.

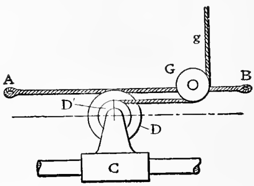


FIG. 114.—Armand Stewart's Reducing Motion.

Reducing Wheels are not infrequently made for attachment directly to the indicator, as illustrated in **Figs. 115** and **116.** The former shows the **Crosby** reducing-wheel attachment and the latter a similar device for the **Tabor** indicator.

Calculations of the Indicated Horse Power of an engine show usually the power developed on **one side** of the piston, which is commonly stated by the formula,

$$\text{I.H.P.} = \frac{\mathbf{plan}}{33{,}000}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (25)$$

Where **p** = mean effective pressure on the piston, lbs. per sq. in.;

l = length of stroke in **feet**;
a = net area of piston in square inches;[1]
n = number of revolutions per minute.

[1] In all piston engines the area of the piston rod must be subtracted from the area of the piston on the side where the rod reduces the area effective for the action of the steam or other working substance.

Of the terms of this equation only one, the mean effective pressure, is obtained from the indicator cards.

If we consider now only **one end** of the cylinder, the steam does work on the piston during a "forward" stroke, and, on the other hand, the **piston does work on the steam** on the "return" stroke. Hence to get the mean effective pressure for a stroke the average pressure during the return stroke must be subtracted from the average pressure on the "forward" stroke; and this is obviously the same as the average length of all the ordinates intercepted between the upper and lower lines of the indicator card multiplied by the scale of the spring.

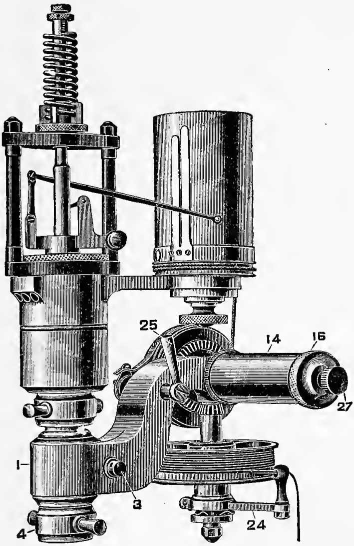


Fig. 115.—Crosby Indicator and Reducing Motion Attachment.

Usually the mean effective pressure is found by means of planimeters, the use of which for this purpose was explained on pages 72–77. An engineer should, however, know how to calculate the mean effective pressure of an indicator diagram with reasonable accuracy without the use of such instruments. In such cases the **method of ordinates** is very convenient. With suitable triangles draw ordinates perpendicular to the atmospheric line at both ends of the diagram as shown in **Fig. 117.** Lay off on the edge **AB** of a piece of smooth flat paper, a scale of ten equal divisions so chosen that the total length of the ten divisions is a little greater than the length of any of the indicator diagrams. This scale should then be placed **obliquely** across the diagram to be measured, so that the beginning and

end of the scale will be located on the ordinates at the ends of the diagram. Now mark the diagram opposite the divisions of the scale with fine points, and at the **middle** of each of these

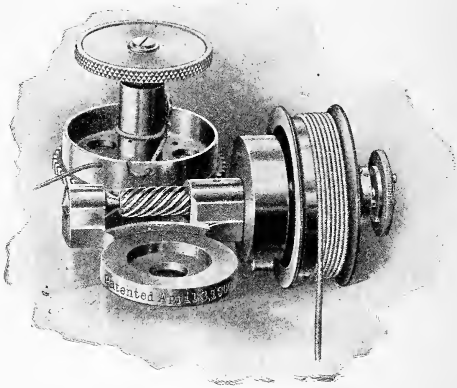


FIG. 116.—Reducing Motion Attachment for Tabor Indicator.

divisions draw ordinates across the breadth of the diagram. The sum of the lengths of these ordinates divided by ten gives the value of the **mean ordinate,** and this when multiplied by

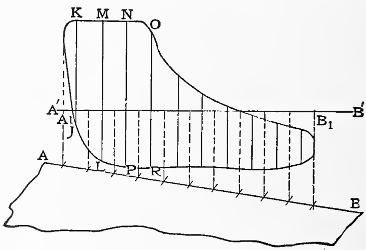


FIG. 117.—Diagram Illustrating Method of Mean Ordinates.

the true scale of the spring gives the **mean effective pressure.** Some time can be saved in summing the ordinates if they are transferred with a dividers one after the other to the edge of the strip of paper. The total length laid off divided by ten is then the mean ordinate.

The Engine Constant for Indicated Horse Power. In the use of equation (25), page 118, where

$$\text{I.H.P.} = \frac{plan}{33{,}000},$$

considerable time can usually be saved when calculating engine tests if the terms

$$\frac{la}{33{,}000}, \quad \ldots \ldots \ldots \quad (26)$$

called the **engine constant,** which always remain constant for **each end** of the cylinder, are first computed carefully and then used as constants throughout the calculations. In other words, the **indicated horse power** is found for each end of the cylinder by taking the product of the terms,

$$\textbf{Engine Constant} \times p \times n.$$

CHAPTER VI

MEASUREMENT OF POWER—DYNAMOMETERS

A DYNAMOMETER, according to its derivation, is an instrument for measuring force or "power." These are of two kinds:

1. Those **absorbing** the power by friction and dissipating it as heat.

2. Those **transmitting** or passing on the power they measure, thus wasting only a small part in friction.

Absorption Dynamometers (Prony Brakes). Of the class of dynamometers in which the power received is all **absorbed**

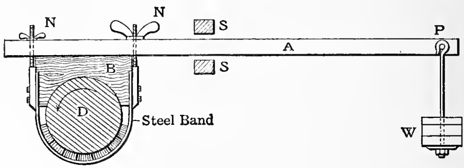


FIG. 118.—Simple Prony Brake.

in friction, the type generally used is called a **Prony brake,** named for Rev. John Prony, who many years ago developed a device of this kind for measuring power.[1] One of the simplest forms is shown in **Fig. 118.**

It consists of a lever **A,** from which a weight **w** is suspended from one end, and a block **B,** supported on a revolving drum or pulley, is attached to the other end. A strap to which wooden cleats are fastened is held in place and tightened by the thumb-nuts **N, N.** When the friction on the strap and block just

[1] Strictly speaking, a brake of this kind does not provide means for directly measuring power, as, for example, horse power, because the element of time is not indicated. In other words, it measures the **tangential force** of which a couple (**torque**) in linear-weight units, such as foot-pounds, can be computed.

balances the weight w, the lever arm **A** is horizontal and the apparatus is in adjustment. Stops, marked **S, S,** are provided so as to limit the movement of the lever arm.

When the brake is adjusted or "balanced," the work done in a given time in producing the friction (the power absorbed) is measured by the weight moved multiplied by the distance it would pass through in that time if free to move. Then if,

r = length of brake arm in **feet.**[1]
n = revolutions of the shaft per minute.
w = weight on the brake arm in pounds.

$$\textbf{Brake Horse-power (B.H.P.)} = \frac{2\pi rnw}{33{,}000}. \quad . \quad . \quad (27)$$

In equation (27) the fraction $\frac{2\pi r}{33{,}000}$ is a constant quantity for a given brake and is called the **brake constant.** When a brake like the one in **Fig. 118** is used the **effective** weight of the brake itself as weighed at the point **P** must be **added** to the weight **w.** A very common variation[2] of the Prony brake is illustrated in **Fig. 119.** Rotation being in the opposite direction from that in **Fig. 118,** the knife-edge at **E** on the arm **A** will now **press** on the pedestal **T,** and the weight **w** can be determined by weighing the pressure on a platform-scales **S.** Since the scales receives not only the pressure due to the force producing friction, but also that due to the weights of the brake and of the pedestal, these weights must be determined and are to be subtracted from all of the readings of the scales to obtain the **net weight w,** for substitution in equation (27). Weight of the brake and the pedestal, called the **zero reading,** must be obtained with the brake strap **slack,** so that the block **B** will rest as lightly as possible on the pulley. With small engines this zero reading is obtained most

[1] The length of the brake arm is measured by the perpendicular distance from the line of action of the weight **w** to the center of the wheel. When the arm **A** is **horizontal,** as in Fig. 118, the length of the brake arm is usually measured by the horizontal distance from **P** to a line passing through the center of rotation perpendicular to the arm.

[2] For information regarding the designing of Prony brakes for absorbing large powers the reader is referred to *Engine and Boiler Trials*, by R. H. Thurston, pages 260–279.

accurately by observing the weights on the scales when the brake pulley is turned around by hand first in one direction and then in the other. In this way we obtain for both the brake and pedestal, with rotation in one direction the weight plus the friction due to their own weight, and with rotation in the other direction the same weight minus the same friction. Half the sum of the two readings is, therefore, the weight corresponding to the pressure on the scales due to gravity alone. With large engines it is sometimes difficult to turn them uniformly by hand, so that the zero reading must be obtained by some other method. This is done usually in prac-

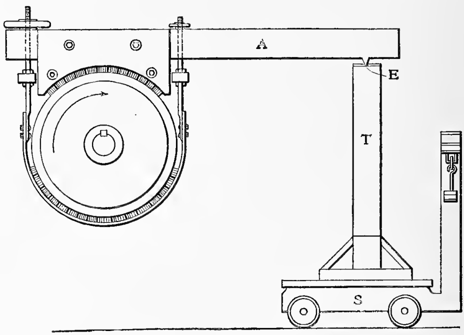


FIG. 119.—Prony Brake with Platform Scales.

tice by placing a very small rod on the drum or pulley **D** (**Figs. 118** and **119**) vertically over the center of the shaft. Then if the strap is loose and due care is observed, the pressure on the scales can be obtained with sufficient accuracy without rotation.

Other forms of Prony brakes are illustrated in **Figs. 120** and **121.** The former is called a **strap brake** and the latter a **rope brake.**

The strap brake is made up merely of a band of steel or leather or of strands of rope placed over or wrapped around a suitable pulley.[1] In this case weights must be suspended from both sides of the brake wheel or pulley.

[1] It is desirable to use for Prony brakes pulleys of which the section of the face is a double "U," like **Fig. 122.** The outside rims are for keeping the brake in position on the pulley and those on the

Rope brakes [1] like the one shown in **Fig. 121,** are much used for "commercial testing" of engines, as it is easily portable or can be made quickly at a small expense from materials always at hand. Moreover, it is self-adjusting, so that accurate fitting is not required. It consists of a rope doubled around a pulley or fly-wheel on the shaft transmitting the power to be measured. Several U-shaped distance pieces of wood, preferably maple, are provided to prevent the rope from slipping off the pulley and to keep the parts of the rope separated. These distance pieces should be attached to the rope by soft iron or copper belt lacing, drawn in from the outside of the wooden pieces through the center of the rope, instead of being fastened with screws or nails on the inside, which will heat to a high temperature and then char the rope. Sometimes such fastenings when of hard metal will cut grooves into the surface of the pulley.

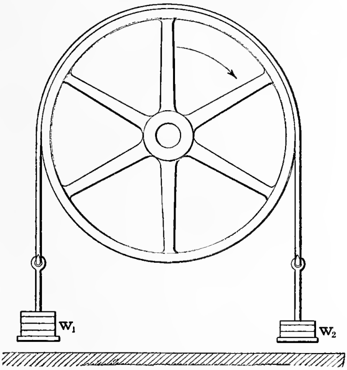


FIG. 120.—Strap Brake.

In the case of the **strap** brake, **Fig. 120,** the net pull, corresponding to the weight **w,** in equation (27), page 123, is $w_1 - w_2$.

inside for receiving a small stream of water played upon the inside of the pulley. This stream of water by its evaporation will assist materially to dissipate the heat generated. Such brakes are often operated with pipes arranged to discharge water into the pulley and another pipe to carry it away. This is an excellent system provided the latter pipe is used only to carry away a little overflow, but if so much water is used that there is practically no "steaming," the inside rim of the pulley will fill up with water to be spattered around in every direction as well as over the face of the pulley, where it is particularly objectionable, as it produces variable friction.

[1] Sir William Thomson (Lord Kelvin) invented in 1872 the first rope brake of which we have any record. Although he utilized this device as a friction brake, it was not used by him as a dynamometer.

Now the same relation would hold if, as is often done, a spring balance is fastened to the floor on say the left-hand side; the pull registered by the spring balance would then be w_1[1], and the net pull is, as before, w_1-w_2. Brake horse power is calculated then by equation (27), substituting for **w** the net pull

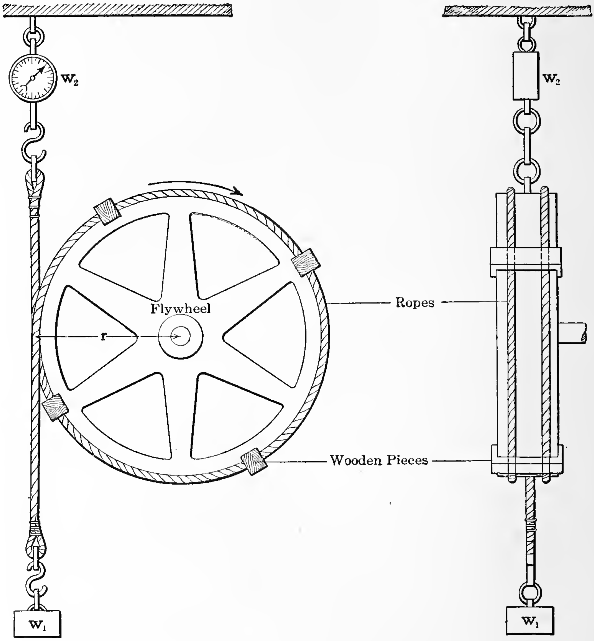


FIG. 121.—Rope Brake.

w_1-w_2. The same is true also for the rope brake in Fig. 121, so that for these two dynamometers we have

$$\text{Brake Horse Power (B.H.P.)} = \frac{2\pi rn(w_1-w_2)}{33{,}000}. \qquad (28)$$

[1] To the pull (w_1) must be added, however, the weight of any hooks placed between the end of the strap or rope and the spring balance; and if the balance is for any reason suspended in the inverted position the weight of the balance itself must also be added.

Where r is the radius of the wheel plus one-half the diameter of the rope in **feet**, and **n** is the revolutions of the shaft per minute.

Alden Brake. An entirely different type of absorption dynamometer is the Alden brake, which is illustrated in **Fig. 123.**

In this apparatus the rubbing surfaces producing the friction necessary for absorbing the power are separated by a film of oil, and the heat generated is carried off by a stream of water circulating through it. It consists of a disk of cast-iron **A,** which is to be connected to the shaft **S,** transmitting the power. This disk revolves between two thin copper plates **E, E,** fastened together at their outer edges to form a shallow cylinder which

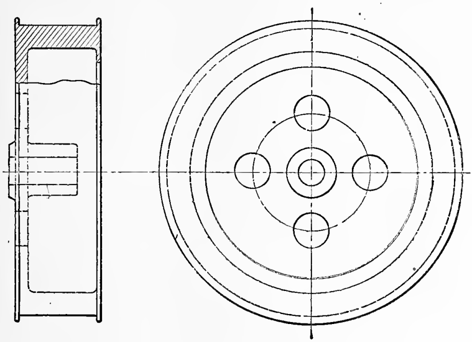


Fig. 122.—Brake Pulley.

is filled with a bath of heavy (cylinder) oil. Water under pressure is discharged into the chamber adjoining the copper, plates and any increase in pressure causes the copper plates to press against the cast-iron plate **A** with more force. The friction between these copper and iron plates tends to turn those of copper, but as these are rigidly connected to the outside casing **C,** carrying the brake arm **P,** the tendency to turn can be determined by weighing as with a Prony brake. To maintain the moment of resistance constant under all circumstances the pressure and consequently the flow of water into the casing is automatically regulated by a cylindrical valve **V,** which becomes partially closed if the brake arm moves above a certain horizontal position. This valve is shown in section

in **Fig. 124.** The end at **W** is connected to the water main and the other end **Y** to the brake casing by means of a right-angled bend **R (Fig. 123).** Water entering by the pipe **W** passes through the ports **N** and then through the ports **H** into the pipe **Y.** Now a small angular movement of the pipe **W,** relative to the pipe **Y,** will open or close the ports **H** and thus regulate the supply of water. These ports are very narrow,

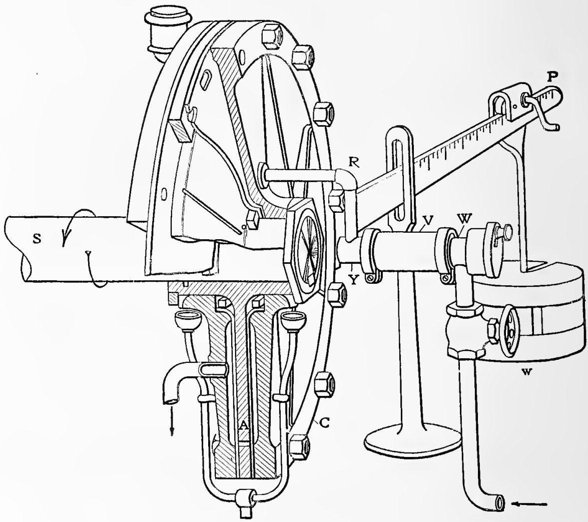


FIG. 123.—Alden Brake.

so that a very small angular motion is sufficient to close them. By making the outer casing of the valve of rubber it is found to be sufficiently flexible to permit moving the valves, and at the same time it offers very little resistance to the movement of the casing.

Water Brakes. Power is absorbed by moving, in water, a rotor similar in most cases to those in steam turbines. Such an apparatus is called a water brake. A good example is shown in **Figs. 125** and **126.**

It is the type used by the Westinghouse Machine Company

of Pittsburg for testing the power of large steam turbines. It consists of a rotor **R,** mounted on a shaft **S, S,** of which one end is arranged to be connected directly by means of a coupling to the shaft of the turbine or engine to be tested. This rotor revolves within a closed casing **Z,** supported on the journals **J, J,** through which the shaft passes. Around the periphery of the rotor there is a series of rows of vanes which when revolving tend to give to any water contained in the casing a

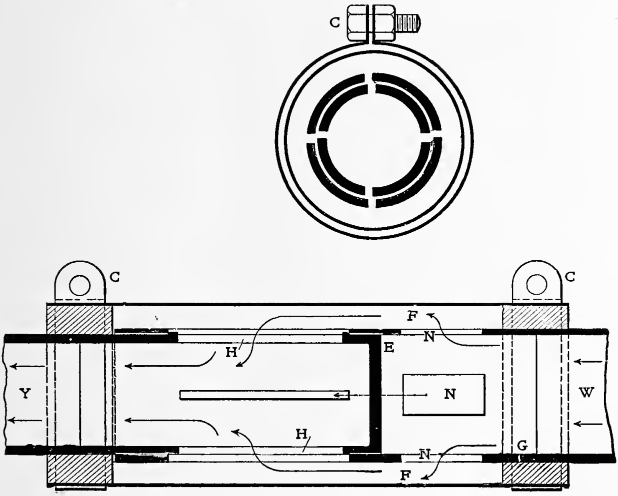


FIG. 124.—Regulating Valve in Alden Brake.

rotary motion. Between every two rows of vanes on the rotor there is fitted a row of stationary vanes attached to the inside of the casing. This arrangement is illustrated diagrammatically in **Fig. 126,** where the cross-hatched sections represent the stationary vanes and the "solid" sections the moving vanes. The tendency of the moving vanes to produce this rotary motion of the water is, as it were, resisted by the stationary ones, and this action develops a very large amount of heating, due to fluid friction in a manner analogous to the operation of a Prony brake or any other absorption dynamometer. As a

result of this friction a force is developed tending to turn the casing in the direction of motion of the rotor. The casing, however, is prevented from turning by a radial arm bearing down on a platform scales. The intensity of this tendency

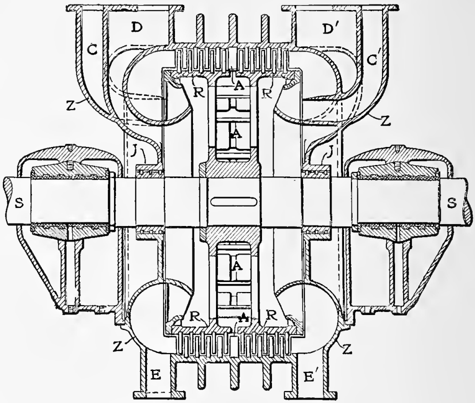


FIG. 125.—Westinghouse Water Brake.

to turn can be regulated as in the Alden brake (page 127) by adjusting the valves controlling the flow of water through the casing. The vanes on the rotor are of the same kind as used in steam turbines and have the function of imparting a high

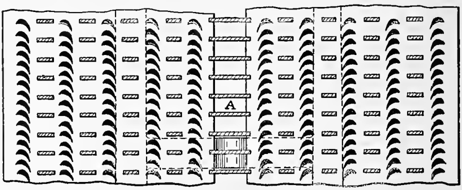


FIG. 126.—Vanes of Westinghouse Water Brake.

velocity to the water flowing through them in an axial direction. Water enters the casing through the inlet pipes **C, C′ (Fig. 125)**, discharging a stream from both sides of the casing toward the central portion of the rotor. At the middle of the periphery of

the rotor there are a number of slots or ports **A, A, A, A** through which the water discharges into the right and left, passing first through a broad central row of stationary vanes shown in **Fig. 126.** Escaping from these vanes it is picked up by the rows of moving vanes on each side, which give it a high velocity and, in turn, discharge it into the adjacent rows of stationary vanes where the velocity just acquired is checked, and so on across the face of the rotor, the moving vanes adding velocity only to be lost in the next row of stationary vanes. From the last rows of moving blades the water is discharged into semicircular passages **B, B′,** which direct the flow of water into the center of the rotor when the cycle is repeated. The water brake illustrated here was designed for high powers, so that a number of rows of vanes was necessary. For small powers of, for example, 200 to 500 horse power, not more than two rows of moving vanes with the corresponding number of stationary vanes would be required.[1]

Power absorbed by a water brake is calculated in the same way as for an ordinary Prony or rope brake.

In the operation of the brake the water is quickly raised to the boiling-point and a considerable portion evaporates, carrying off as steam very large quantities of heat. Vents for the escape of this steam are indicated in the figure, showing the cross-section of the brake by **D** and **D′**. Unless considerably more water, however, was admitted to the casing than was required to replace that lost by evaporation, the action of the brake would be more or less irregular, so that an excess of water is supplied and there is a constant discharge of **hot water** through the passages marked **E** and **E′**.

Dynamos (Electric Generators and Motors) as Power Dynamometers. One of the most convenient means for measuring the power of high-speed engines and turbines is to connect an electric generator to the main shaft. Then if the efficiency of the generator is known at the particular speed and output at which it is to be operated, a very accurate method of measuring the power of the engine or of any other type of motor becomes readily available. The output of the generator should be determined by observations of the volts and

[1] To calculate the resistance of such vanes see "The Steam Turbine," by the author, pages 115–125.

amperes with carefully calibrated **portable** instruments. Remembering that for direct-current generators volts times amperes gives watts and that 746 watts are equivalent to one horse power, then if **E.H.P.** is the horse-power output of the generator we have,

$$\mathbf{E.H.P.} = \frac{\text{volts} \times \text{amperes}}{746}.$$

It is not unusual to hear this result called the "**Electrical**" **horse power.**[1]

The actual horse power delivered to the generator is, of course, the **brake horse power,** and into this result the efficiency of the generator enters. Thus,

$$\mathbf{B.H.P.} = \frac{\text{volts} \times \text{amperes}}{746 \times \text{efficiency of generator}}.$$

As a rule, however, electric **motors** are more serviceable in mechanical engineering laboratories as power dynamometers, because the efficiency is much more easily obtained, the usual method being to determine an efficiency curve for varying power inputs by a Prony brake test, stating this efficiency as,

$$\text{Efficiency of motor} = \frac{\text{B.H.P.}}{\text{E.H.P.}},$$

where E.H.P. is the "electrical" horse-power input, as measured with voltmeter and ammeter.

If then a pump, air compressor, ventilating fan or a similar machine is to be tested by the electrical method, it should be direct-connected to the shaft of the motor, and its efficiency, **E**, will be

$$\mathbf{E} = \frac{\text{U.H.P.}}{\text{E.H.P} \times \text{efficiency of motor}},$$

where **U.H.P.** is the useful work done by the machine, in horse power.

[1] When an alternating-current generator is used the power factor must also be measured with a suitable instrument. In this case, the actual E.H.P. is (volts × amperes × power factor) ÷ 746. This method is discussed more in detail on pages 293–295.

This last method serves also as a convenient method for obtaining the efficiency of a generator, since by connecting it directly to the shaft of a motor previously "calibrated" (for efficiency) the electrical output of the generator and the input to the motor are readily determined.

When a so-called **variable speed** motor is used as a dynamometer, its efficiency must be determined at the particular speed and power at which it will operate when driving the machine to be tested.

Transmission Dynamometers. Instruments of this type are used to measure the amount of power transmitted without absorbing any more power than is absolutely needed to move the dynamometer.

Goss Belt Dynamometer. One of the simplest forms of transmission dynamometers, designed by Professor W. F. M. Goss, is illustrated in **Fig. 127** by a line drawing. Being so simple in construction, it can readily be made in any factory or workshop with the materials available. It is essentially a **differential lever** measuring the difference in tension between the two sides of a belt. This lever is pivoted at the point, **D,** and to it are attached the shafts carrying the pulleys **A** and **B.** Power transmitted is measured by the product of the speed of the belt and the difference in its tension between the two sides of the dynamometer.

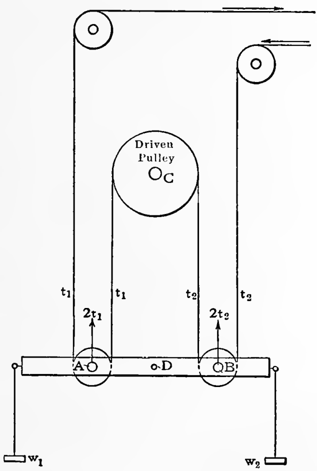


FIG 127.—Goss Belt Dynamoneter.

The force tending to raise the left-hand end of the lever is assumed to be twice the tension t_1, of the tight side of the belt, while that raising the right-hand side is twice the tension t_2 in the slack side of the belt. These tensions on the two sides of the lever can be measured by weights w_1 and w_2 suspended from its ends. The force tending to rotate the lever is therefore

twice the difference in tension on the two sides of the belt $2(t_1-t_2)$ and this force acts with a leverage **AD** = **DB**, when these two arms are made equal. If now the distance from the pivot **D** to the point of support of the weight w_1 is made twice **AD**, and if the weight w_1 is made equal to the difference in the tensions it will balance the lever. Or for these conditions we can write the equation,

$$w_1 = t_1 - t_2.$$

Now in any transmission system the difference in the tension of a belt or a rope on the two sides of a driven pulley multiplied by its speed is a measure of its power. If then, **d** is the diameter of the driven pulley **C**, plus the thickness of the belt or rope in feet, **n** is the number of revolutions of this pulley, and w_1 is the weight in pounds on the left-hand side required to **balance** the lever; when power is transmitted, then,

$$\textbf{Horse power transmitted} = \frac{\pi d n w_1}{33{,}000}. \quad . \quad . \quad . \quad (29)$$

To reduce the vibrations of the apparatus a dash-pot is connected to the right-hand side and also to prevent excessive movement of the lever when unbalanced, dash-pots are placed above and below the lever on the left-hand side.

Differential Dynamometers. The apparatus illustrated in **Fig. 128** is typical of a number of dynamometers indicating by means of a differential lever operated by gearing, the amount of power transmitted.[1] This is a very common form of **transmission dynamometer.** Power is received from the motor (or engine) by the shaft **A**, which is connected only indirectly by means of gears to the shaft **A′** opposite, which transmits the power to the work. To the adjoining ends of these shafts bevel wheels **B** and **D** are attached. The lever **L** turns on an axis **concentric** with the shafts **A** and **A′**, in a plane perpendicular to

[1] Similar forms of differential dynamometers are known as White's, King's and Bachelder's. The first instrument of this kind, it is stated, was invented by Samuel White in 1780. These dynamometers are sometimes called **epicyclic**, signifying "wheels traveling around a circle or around another wheel."

them. It carries bevel wheels **C** and **C_1**, gearing with **B** and **D**, through which the power is transmitted. There is a tendency then for the left-hand end of **L** to go downward and for the right-hand end to rise. If, furthermore, the lever **L** were permitted to revolve, no work would be transmitted from **B** to **D**, and there-

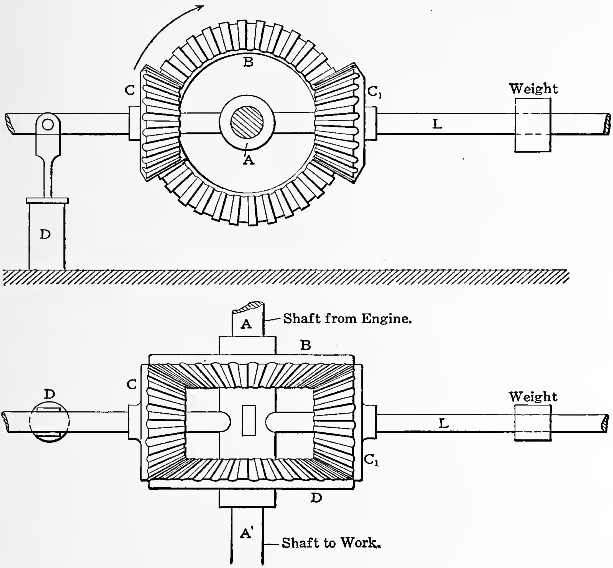


FIG. 128.—Typical Differential Lever Dynamometer.

fore **D** would remain stationary. As these gears are usually proportioned so that **B** revolves with twice as many revolutions in a given time as **L**, a weight placed on **B** at a given radius from the center will balance a weight twice as large at the same radius on the lever **L**. Then the moment of the force applied to the lever **L** to balance it must be twice as great as the moment of the force transmitted from **C** and **C_1** to **D**. If, therefore, **l** is the length (feet) of the arm at which the weight **w** (pounds)

is applied, and **n** is the number of revolutions per minute of the shaft **A**, then the POWER TRANSMITTED PER MINUTE =

$$\frac{2\pi lnw}{2} \text{ (foot-pounds)}, \quad . \quad . \quad . \quad . \quad . \quad . \qquad (30)$$

and

$$\textbf{Horse power (H.P.)} = \frac{\pi lnw}{33,000}. \quad . \quad . \quad . \quad . \quad . \qquad (31)$$

Now if the lever **L** is to be prevented from turning about its axis, **a couple will be required which is twice the driving couple being transmitted.** If, then, a weight of **w** pounds sliding on **L** as shown in the figure is placed at a distance **l** feet from the axis, so that the lever remains horizontal, the driving couple can be determined; and when the revolutions of the shaft **A** are known, the power can be found. A dash-pot **D** is usually attached to the differential lever **L** to reduce vibrations. This dash-pot should always be kept filled with glycerine or good clean oil. If the dash-pot is sticky consistent results cannot be expected.

A Webber Differential Transmission Dynamometer as made commercially is illustrated in **Fig. 129.** The scale on the lever arm of this instrument is graduated into 100 divisions and a bell is provided which rings at every 100 revolutions. Since the horse power transmitted in one **revolution** per minute is $\frac{\pi lnw}{33,000}$ equation (31), then the horse power corresponding to one division on the scale per 100 revolutions per minute is also $\frac{\pi lw}{33,000}$ for a perfect calibration.

It is interesting to observe that if we let

- **v** = the vertical force acting at **C** and $\mathbf{C_1}$;
- **p** = the vertical pressure between the teeth at each point of contact;
- **d** = distance from the center of the rotation to **C** and $\mathbf{C_1}$;
- **l** = the distance from the same center to the weight **w.**

Then from the foregoing discussion it should be clear that $2v = 4p$ and $wl = 2vd = 4pd$. Then if **r** is the effective pitch radius of the driving gear wheel **B,** r_1 is the radius of the small

bevel wheels, and the force producing the turning movement in the shaft **A** is represented by **f,** we have,

$$\mathbf{fr} = 2\mathbf{pr}_1,$$

and

$$\mathbf{f} = \frac{2\mathbf{pr}_1}{\mathbf{r}} = \frac{\mathbf{wlr}_1}{2\mathbf{dr}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (32)$$

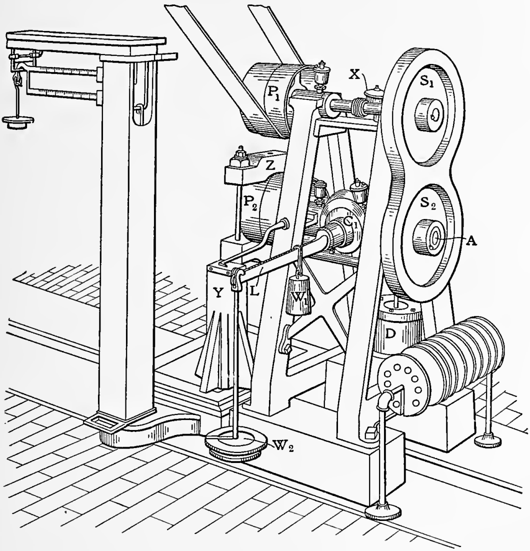


Fig. 129.—Webber Differential Transmission Dynamometer.

If we know the number of revolutions, then the space passed through by the force **f** can be calculated, and the work in foot-pounds is the product of the force times the distance passed through. The units given above are of course respectively in feet and pounds.

Calibration of a Differential Dynamometer. 1. Examine the dash-pot and observe whether the piston moves freely in the cylinder, particularly without "sticking." After the apparatus has been well oiled the position of the poise to make

the lever arm horizontal should be observed for "no load." If this is not at zero then all the readings on the scale must be corrected by the amount of this zero reading.

2. At each of the speeds required make a preliminary run without load and observe the reading of the poise when the lever is balanced.

3. Attach a Prony brake to the shaft from which the power is to be transmitted and observe for a series of loads and speeds the readings of the poise on the dynamometer lever.

4. For each speed plot a curve with theoretical **foot-pounds per minute** by equation (30) as abscissas and actual foot-pounds per minute as determined by the Prony brake as ordinates.

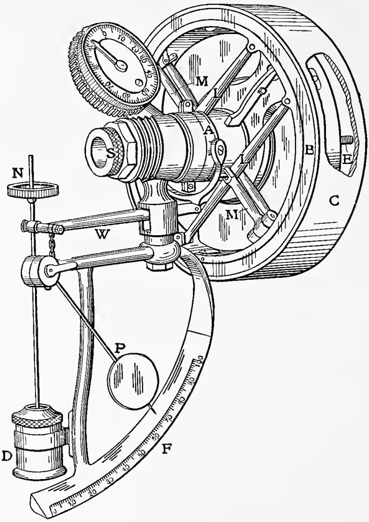


Fig. 130.—Emerson Power Scales.

Emerson Power Scales. Another very satisfactory instrument for the measuring of power transmitted by shafting is known as the Emerson Power Scales. It is illustrated in **Fig. 130.**

It consists of a pulley **C** keyed to the shaft. To this pulley **C** a wheel **B** is connected loosely **by studs EE projecting between and bearing against its spokes.** The pressure exerted on these studs is proportional to the power transmitted by means of the pulley **C,** and this pressure is transmitted by a system of levers **LL** and bell-cranks **MM** to a sleeve **A** connected to a "weighing lever" **W.** The sleeve slides loosely on the main shaft. The amount of the pressure exerted on the studs is indicated for small values by a pointer **P,** moving over a graduated scale

F. For pressures beyond the limits of the graduated scale weights are placed on the scale pan **N.** A dash-pot **D** is provided to prevent excessive vibrations and make the pointer "dead-beat."

The scale **F** is calibrated to read the pressure (**force**) exerted by the torque of the pulleys **C** on the studs **EE** in **pounds.** The

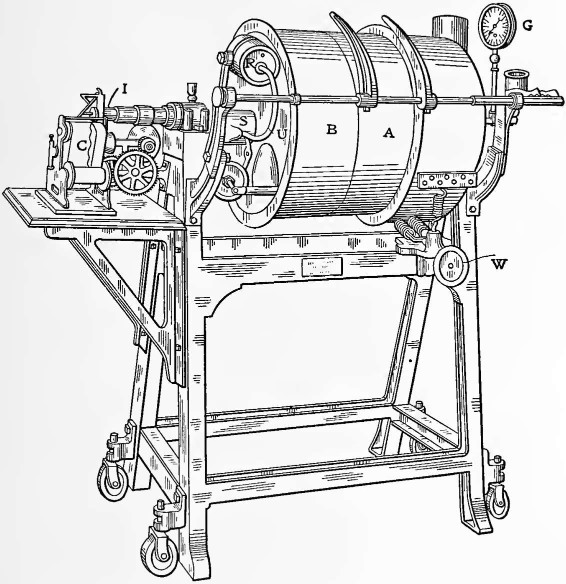


FIG. 131.—Flather's Hydraulic Transmission Dynamometer.

work or "power" is calculated, therefore, by taking the product of this force times the distance moved through. If **d** is the diametral distance between the centers of the studs in feet, **n** the revolutions per minute and **w** the reading of the scales, then,

$$\textbf{Horse power}^1 = \frac{\pi \mathbf{dnw}}{33{,}000}. \quad . \quad . \quad . \quad . \quad . \quad . \qquad (33)$$

[1] Compare with (31) for differential dynamometers, page 136.

A speed counter is attached to the apparatus for counting the number of revolutions.

Flather's Hydraulic Transmission Dynamometers. A form of transmission dynamometer which is operated by **hydraulic**

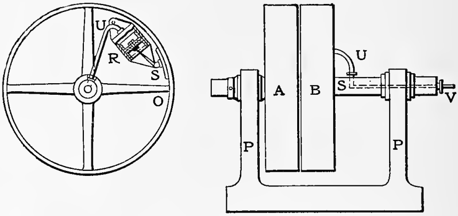


FIG. 132.—Diagram Showing Pulleys, Pistons, and Shaft of Flather's Dynamometer.

pressure is shown in **Fig. 131.** The power shaft is keyed to the boss of a pulley **B** with two or more arms carrying hydraulic cylinders **R.** Projecting ends or studs from these cylinders bear upon the arms of a loose pulley **A** on the same shaft. The torque

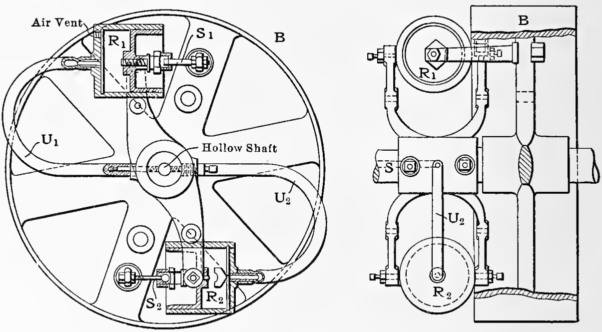


FIG. 133.—Details of Pistons and Cylinders in Flather's Dynamometer.

imparted by the driving belt to the loose pulley **A** is thus transmitted to the shaft **S** through the liquid, and the resulting pressure is conveyed by radial pipes **U** to the hollow central shaft, and then to a pressure gage **G.** The hollow shaft is always filled with oil. In the figure an engine indicator **I** is

shown attached to the hollow shaft for recording the pressure. The loose pulley **A** drives the tight pulley **B** through its pistons which press on the oil in the cylinders carried by the tight pulley. By means of a worm drive the drum of the indicator receives its motion from the central shaft **S.** **Figs. 132** and **133** show more in detail the construction of the hydraulic cylinders on the pulley **B.** **Fig. 134** shows typical indicator

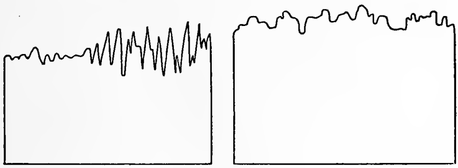


FIG. 134.—Indicator Diagrams from Flather's Dynamometer Attached to a Mining Drill.

diagrams from this apparatus. Both were taken from a dynamometer connected to a mining drill. The first was taken when the drill was sharp, the second when it was dull.

Among the advantages claimed are: (1) its simplicity, (2) that it is not appreciably affected by the velocity of the shafting, (3) that no countershaft is required, (4) by connecting it to a recording gage a continuous diagram of the load can be obtained.

CHAPTER VII

FLOW OF FLUIDS

THE flow of fluids will be discussed under these heads:

1. The flow of air.
2. The flow of steam.
3. The flow of water.

The Flow of Air. When subjected to only a low pressure, air and many other gases are usually measured by a **gas meter,** of which there are many types sold commercially. There are, however, two general types: (1) **wet** and (2) **dry.** The former is by far the more reliable and should be always used in preference to a dry meter when it can be obtained. Wet meters receive their name from the water seal maintained in them. This seal must be always kept at a constant level, determined by calibration,[1] and before using such a meter in a test one should always observe whether the water level is at the standard mark. If it is not, then water must be added or withdrawn as the case may be. A section of a wet meter is shown in **Fig. 135.** It consists, as usually made, of a series of chambers arranged like an Archimedean screw, which are alternately filled and emptied. When air or any gas flows[2] into one of the chambers of the meter it accumulates over the surface of the water and by its pressure raises the chamber until it is filled. In the figure the gas enters at the dry-well, **V,** passes through the drum and out at the front end, then over the drum between it and the case to the outlet. In this way the drum is made to revolve to the left by the pressure on the

[1] Gas meters may be calibrated by any apparatus suitable for the displacement of gas as it is withdrawn by water. It is very necessary, of course, that when the weighings are made the pressure and temperature of the gas be accurately determined.

[2] The nature of specific gravity of the gas is not important, as gas meters are calibrated to record volumes, usually cubic feet.

surface of the water below and the slanted partition **C** above forming an ever-increasing pyramidal space between the surface of the water and the plane of the slanted partition. Wet

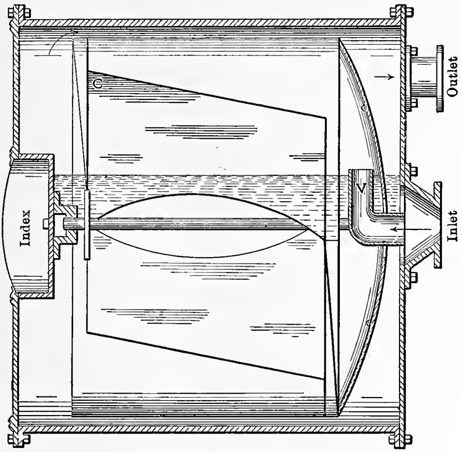


FIG. 135.—Typical Wet Gas Meter.

meters are usually very accurate, while dry meters are not "supposed to be instruments of great accuracy."

Pitot Tube for Measurement of Air. Probably the most accurate method of measuring air in large volumes is by means of a Pitot tube. A standard instrument of this kind designed for the measurement of air by the American Blower Company is shown in **Fig. 136.**

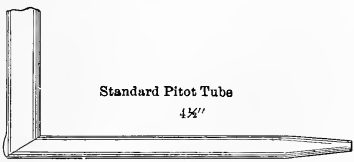


FIG. 136.—"American Blower Co." Standard Pitot Tube.

It consists simply of two tubes, a small one, being placed inside of a larger one as illustrated in detail in **Fig. 137.** These tubes are arranged so that each has a separate connection,

as at **A′** and **B′**. The lower end of the small tube is open at **A**, while the outside and larger tube has two openings at the opposite sides of **B** marked in **Fig. 137**, $2\frac{3}{16}$ inches from the end at **A**. When the instrument is used it is placed so that the opening at **A** points against the direction of flow and receives the full effect of the pressure due to the velocity of flow. The side openings in **B** are subjected to only the static pressure. For convenience let **p** = velocity pressure and **s** = static pressure. For example, the difference in the levels in the manometer, **a**, **Fig. 138**, is therefore that due to (**p**+**s**) − **s**, or simply **p**, the **velocity pressure**.

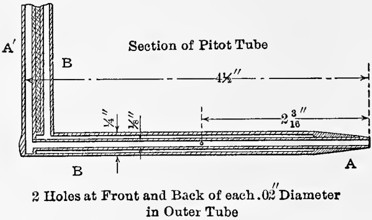


FIG. 137.—Section of Standard Pitot Tube.

Pitot tubes are usually connected to manometers or preferably to sensitive draft gages, showing the pressure in small fractions of an inch of water. When the end of the Pitot tube

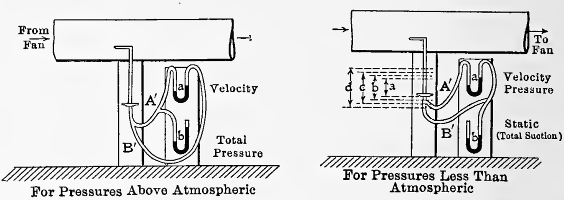


FIG. 138.—Arrangement of Connections for Pitot Tube Measurements.

at **A′** is connected to the left-hand end of a draft gage like those in the figures on page 24 and the end at **B′** is attached to the right-hand end, the instrument acts as a differential gage and the difference between the reading when thus connected and its zero reading is the pressure in inches of water corresponding to the velocity alone; that is, (**p**+**s**) − **s**. If we call this velocity pressure **P** when reduced to **feet** of water, and if **h** is the height or " head " also in **feet** of an equivalent column

of air producing the same pressure, then the velocity of the air v in feet per second is

$$v = \sqrt{2gh},$$

where g is the force of gravity (32.2), and

$$h = P \times \frac{\text{wt. of a cu. ft. of water}}{\text{wt. of a cu. ft. of air}}$$

$$= \frac{62.3^{1} P}{\text{wt. cu. ft. air}}$$

$$v = 63.3\sqrt{\frac{P}{\text{wt. cu. ft. air}}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (34)$$

In the following table the weight is given of dry air and also the weight of the dry air in a cubic foot of air completely saturated with moisture (100 per cent humidity). The data given are at atmospheric pressure (14.7 pounds per square inch). By interpolating between these tables, the weight of air for any temperature and degree of saturation is easily obtained. Remembering also that the weight per cubic foot is directly proportional to the absolute pressure, the weight for any pressure is readily determined. Tables for determining the percentage of saturation by means of wet- and dry-bulb thermometers are given on page 331.

Temperature, Deg. Fahr.	Weight of 1 Cu. Ft. of Dry Air, Lbs.	Weight of Dry Air in 100 % Saturated Air	Temperature, Deg. Fahr.	Weight of 1 Cu. Ft. of Dry Air, Lbs.	Weight of Dry Air in 100 % Saturated Air
0	.08635	.08623	75	.07424	.07194
10	.08451	.08430	80	.07355	.07090
20	.08275	.08243	85	.07288	.06977
32	.08075	.08024	90	.07222	.06861
40	.07944	.07874	95	.07157	.06773
45	.07865	.07780	100	.07093	.06613
50	.07788	.07688	110	.06968	.06310
55	.07712	.07592	120	.06848	.06025
60	.07638	.07496	150	.06511	.04790
65	.07566	.07396	200	.06018	.01200
70	.07494	.07298			

[1] The weight of a cubic foot of water at about "room" temperature (about 70° Fahr.).

Flow of Air through an Orifice. Air under comparatively high pressures is usually measured in practice by means of pressure and temperature observations made on the two sides of an orifice. **Fig. 139** illustrates the method with two pressure gages on opposite sides of the orifice and a thermometer for obtaining the temperature t_1, at the initial or higher pressure, p_1. The flow of air **w**, in pounds per second may then be calculated by **Fliegner's formulas.**[1]

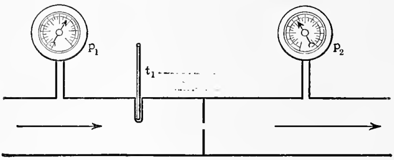


FIG. 139.—Measuring Flow of Air through an Orifice.

$$w = .530 \times a \frac{p_1}{\sqrt{T_1}} \text{ when } p_1 \text{ is greater than } 2p_a \quad . \quad . \quad (35)$$

$$w = 1.060 \times a \sqrt{\frac{p_a(p_1 - p_a)}{T_1}} \text{ when } p_1 \text{ is less than } 2p_a, \quad . \quad (36)$$

where **a** is the area of the orifice in **square inches,** T_1 is the absolute initial temperature in degrees Fahrenheit at the absolute pressure p_1 in the "reservoir," and p_a is the absolute **atmospheric** pressure, both in pounds per **square inch.**

For small pressures it is often desirable to substitute manometers for pressure gages.

Flow of Air Measured by Cooling. This method depends on taking from the air an amount of heat[2] which can be measured and then computing from the heat units absorbed and the difference in temperature and specific heat of the air, its weight and volume. The arrangement of the apparatus is shown in **Fig. 140.** A coil of pipes **C** of which the cooling surface is as equally as possible distributed over the section of the duct **D, D′**

[1] See Peabody's *Thermodynamics*, page 135, and Spangler's *Applied Thermodynamics*, pages 12–13.

[2] The method will be equally applicable if heat is added, as for example by passing steam through the coil. This method is often used to calibrate Pitot tubes and anemometers.

carrying the air to be measured, is used to absorb heat by circulating water through it. Thermometers are arranged so that the temperatures of the air and of the water can be observed, and a platform scales is shown for obtaining the weight of water. Using the symbols t_1 and t_2 for the initial and final temperatures of the air in degrees Fahrenheit, t' and t'' for the temperatures of the water entering and leaving in degrees Fahrenheit, w_a = weight of air passing through D, D′ in pounds per second, w_0 = weight of water collected in pounds per second, and 0.2375 = specific heat of the air at con-

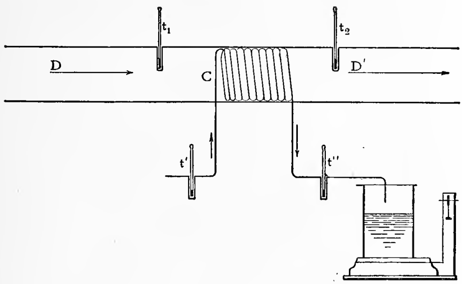


Fig. 140.—Measuring Flow of Air by Cooling.

stant pressure and at temperatures not much above "atmospheric,"[1] then the heat absorbed by the water per second is $w_0\ (t''-t')$ B.T.U. and this equals the heat lost by the air, or $0.2375\ w_a\ (t_1-t_2)$, and

$$w_a = \frac{w_0(t''-t')}{0.2375(t_1-t_2)} = \frac{4.211 w_0(t''-t')}{(t_1-t_2)}. \quad . \quad . \quad (37)$$

Anemometers. A very convenient and simple method for measuring directly the volume of the air, or any gases, is by using an instrument called an anemometer. This instrument, **Fig. 141**, consists in its essential parts of a wheel having flat

[1] The weight of a cubic foot of air is .0765 pound at 62 (522 abs.) degrees Fahrenheit and 14.7 pounds per square inch pressure. Since the volume is directly proportional to the absolute temperature, the weight at any other temperature is easily computed.

or hemispherical vanes mounted on slender arms. The wheel must be made very light in weight, must be accurately balanced, and should move easily in its bearings. By its own motion it operates a recording mechanism indicating velocities in feet per **minute** (in English units).

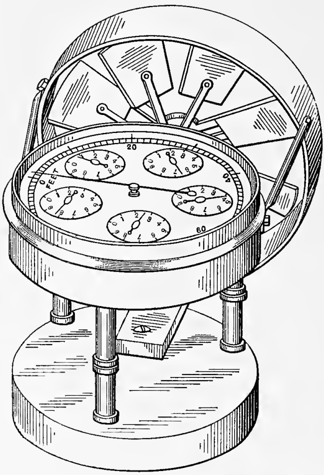


FIG. 141.—A Typical Anemometer for Measuring Velocity of Air.

The Flow of Steam. The flow of the steam from an orifice or nozzle has a very definite critical value when the final pressure is greater than 0.58 of the initial pressure. When the final pressure is less than this critical value the flow is expressed very accurately by the following empirical formula, based on the experiments of Messrs. Emswiler and Fessenden, in the Mechanical Laboratories of the University of Michigan. Using the following symbols:

p_1 = initial absolute pressure of the steam in pounds per square inch;

p_2 = final absolute pressure of steam pounds per square inch;

a = area of the smallest section of the nozzle or orifice in square inches.

Then the weight of the dry saturated steam discharged in pounds per second is approximately,[1]

$$w = \frac{p_1^{.97}a}{60.5} \text{ when } p_2 \text{ is less than } 0.58p_1. \quad . \quad . \quad (38)$$

[1] A somewhat simpler formula, known as **Napier's formula**, which is accurate enough for most calculations, is the following:

$$w = \frac{p_1 a}{70} \text{ when } p_2 \text{ is less than } 0.58p_1. \quad . \quad . \quad . \quad . \quad . \quad (39)$$

Now since in the theoretical formulas the weight discharged is inversely proportional to the square root of the specific volume **v**, or **w** is proportional to $\sqrt{\frac{p_1}{v}}$ the formula above corrected for initial quality **x** of the steam is,

$$w = \frac{p_1^{.97}a}{60.5\sqrt{x}} \text{ when } p_2 \text{ is less than } 0.58\,p_1. \quad . \quad (40)$$

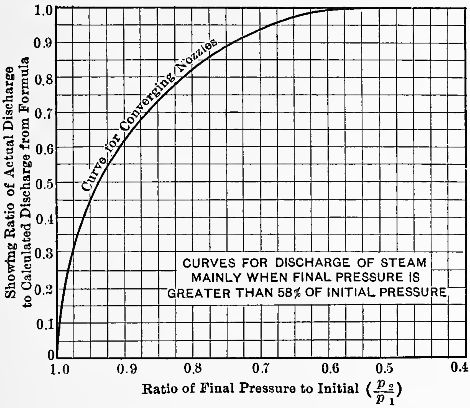


FIG. 142.—Rateau's Curve for Flow of Steam giving Values of the Coefficient **K**.

When the steam is **superheated** the specific volume is considerably increased, and for this condition the author has found that the following equation gives very satisfactory results,[1]

$$w = \frac{p_1^{.97}a}{60.5(1 + .00065d)}, \quad . \quad . \quad . \quad . \quad . \quad . \quad (41)$$

when, as before, p_2 is less than $0.58p_1$ and where **d** is the number of degrees (Fahrenheit) of superheat.

When the final pressure p_2 is greater than $0.58p_1$, the formulas must be modified to correspond to the reduced flow observed by inserting a coefficient **K** as a factor in the right-hand member

[1] For a more extended discussion of the flow of steam see *The Steam Turbine*, by the author, pages 52–57.

of the equations. Values of this coefficient are most conveniently obtained from the curve in **Fig. 142**, which was plotted from the experimental results obtained by Professor Rateau of Paris.

The Flow of Water. When the quantity of water to be measured is not too large it is most accurately determined by weighing in tanks placed on scales, or by direct measurement of volume in calibrated tanks or barrels. Sometimes it is impracticable to weigh or measure the volume of the water directly, particularly when it must be measured **under pressure.** For measurements in pipes up to 2 to 3 in diameter a **water meter** is generally used.

A great many types of water meters are sold commercially and not very many are accurate, so that it is absolutely necessary when using a meter to measure water in a test to calibrate it at least before and after the test, under the same conditions of temperature, pressure, and rate of flow of the water. In many plants where meters are used constantly, suitable connections are made to the discharge from the meter, so that at any time the flow through it can be diverted into a tank in which it can be measured by volume or weighed.

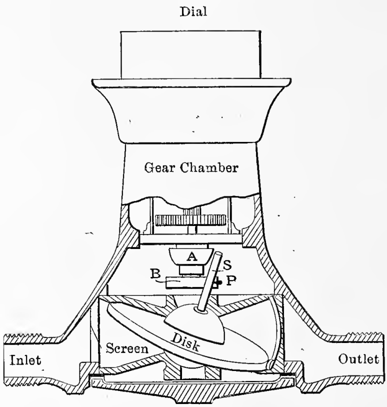


FIG. 143.—Pulsating Diaphragm Water Meter.

One of the best types of water meters is illustrated in **Fig. 143.** This belongs to the class operating with a "pulsating diaphragm." The inclined shaft **S** on this diaphragm traveling around in contact with the peg **P** on the plate **B** moves the counting mechanism through intermediate gears. This diaphragm, in the **Thomson-Lambert meter (Fig. 144)** is made of hard rubber reinforced with a steel plate, making it

much more durable than those made without reinforcing. As the side chambers are alternately filled and emptied, the diaphragm is moved up and down with a kind of "pulsating" motion. A central spindle on the diaphragm is connected to a set of gear wheels operating the recording mechanism.

FIG. 144.—Thomson-Lambert Water Meter.

Worthington Water Meter. The Worthington water meter (**Fig. 145**) is also used frequently. It belongs to the type operating in a cylinder by a reciprocating piston which is driven backward and forward by the pressure of the water. Friction is an important element in meters of this type; but it is not injured by moderately hot water.

The readings of a water meter are usually in cubic feet.

A water meter is essentially a water motor adapted for operating the gearing connected to the counting mechanism.

Frequent calibrations of water meters are necessary because they are likely to become more or less clogged with dirt and refuse. The readings are also affected by the temperature, head, and quantity of water flowing, as well as by the amount of air carried in the water. A meter should always be calibrated at least at two or three rates of flow, as it scarcely ever happens that the conditions of the test are so uniform that the meter will be used only for a certain predetermined rate of flow.[1]

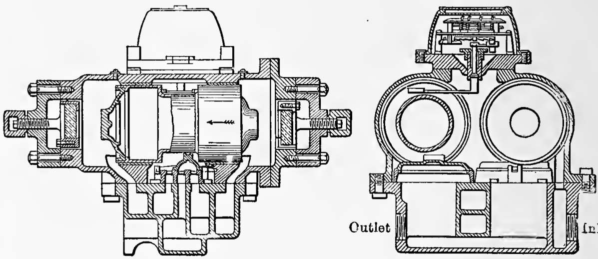


Fig. 145.—Sections of a Worthington Water Meter.

Willcox Water Weigher. Automatic weighing or measuring devices are often used for determining the weight of condensed steam in engine tests or the weight of feed water in boiler trials. The Willcox weigher is a most satisfactory apparatus of this kind. It consists of a tank (**Fig. 146**) divided by a partition **P** into two compartments **A** and **B**, one above the other. The upper compartment **A** receives the inflow of water and the lower one **B** serves for measuring. Projecting into the lower compartment is a U-shaped discharge pipe **C**, which is always water-sealed. The upper end of the discharge pipe is covered by a bell float **F**, which is permitted a short up-and-down movement. In the upper compartment there is a short standpipe **S**, which is simply a hollow cylinder open at the top

[1] Calibration curves are usually plotted with meter readings as abscissas and actual volumes as ordinates. A curve should be plotted for each of the several rates of flow if they are different. Curves of meter readings (abscissas) and correction factors (ordinates) are also useful.

and bottom. The bell float **F** and the standpipe **S** are connected rigidly by a vertical rod (**Fig. 147**) so that they move together as one piece, and this is the only moving part in the apparatus. The lower end of this standpipe has a corrugated face, and when it is down in its lowest position its corrugated face rests on a soft seat or ring surrounding a circular opening in the partition **P.** This seat is made of a rubber composition which is not injured by boiling water. The weigher can be used, therefore, with either hot or cold water without risk.

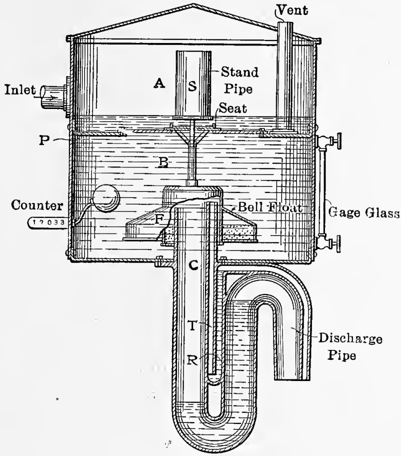


Fig. 146.—Willcox Water Weigher (by Volume).

In the operation of the apparatus, when the standpipe **S** is down on its seat, water entering through the side inlet accumulates in the upper compartment **A** until it overflows the top of the standpipe. The water then flows down through the hollow standpipe into the lower compartment until there is a sufficient amount to seal the lower edge of the bell float **F.** Then as more water accumulates the bell float rises, lifting the standpipe **S** from its seat and the water in the upper compartment flows down into the lower one until the volume is that of a "unit charge" for the apparatus, when the "tripping" device discharges the water through the discharge pipe **C.** The "tripping" is accomplished by a "trip" pipe **T,** which is normally water-sealed, but which becomes unsealed when a "unit charge" has accumulated. While the water is accumulating in the lower compartment **B** the water in the left-hand leg of the "trip" pipe **T** is being slowly pushed down because

of the increasing pressure of the air under the bell float **F**, due to the increase of head of water, and a corresponding amount of water spills over the upper end of the right-hand leg **R** of the "trip" pipe into the discharge pipe **C**. Due to this action the water level in **T** is lowered until it reaches the bend in the lower end of the "trip" pipe. Under these conditions the water column in **R** exactly balances the head of water in the lower compartment **B** and the air entrapped in the float valve **F** has a function similar to that of a scale beam, balancing on one side the head of water in the tank and on the other side the head of the standard water column in **R**, which of course is always constant. At the instant this balance is secured a very small amount of water added in the lower compartment and the corresponding additional spill from **R** will destroy this equilibrium. Then the air compressed in the bell float **F** and the upper part of the discharge pipe breaks the water seal in **R** by suddenly discharging all the water in it. When the air pressure in the bell float **F** is thus reduced it drops down, carrying down with it the standpipe **S** in the upper compartment **A**. In this last operation the air is removed from the interior of the bell float **F**, and water flowing in to replace it will spill over the top of the discharge pipe **C** and will flow out at the other end till the lower chamber is emptied of the "unit charge." At this time, the standpipe **S** becomes seated, due to the pressure of the water above the bell float and to the downward suction of the syphon. Thus the standpipe is held tightly upon its seat only at the instant when tightness is required; that is, while the "unit charge" is being discharged. After the standpipe has seated water again accumulates in the upper compartment **A** and the cycle of operations is repeated.

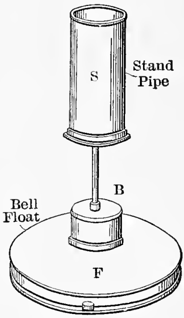


Fig. 147.

A mechanical counter shown at the side of the apparatus is connected to a ball float inside the lower compartment and registers the number of times the apparatus is tripped. An

automatic weigher of this kind is easily calibrated by weighing several "unit charges," and it can then be used with as great a degree of accuracy as can be expected with rapid weighings in tanks on platform scales. The Willcox weigher may be expected to weigh hot or cold water with a maximum error of not more than one per cent.

Venturi Meter. An arrangement of piping in which there is a gradual narrowing of the section to a minimum and then a more gradual enlargement was invented by Mr. Clemens Herschel for measuring the flow of water. This apparatus is called a venturi meter and is shown in **Fig. 148.** Piesometer tubes (manometers) are arranged to indicate the pressure at the sections shown. Pressures at these sections will be denoted respectively by p_m and p_n.

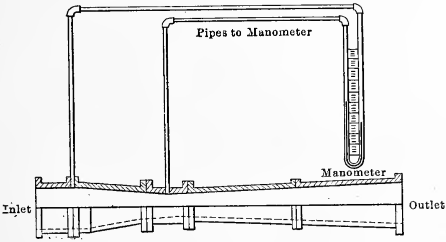


FIG. 148.—Herschel's Venturi Meter.

From Bernouilli's theorem[1] it follows that the relation between the pressure in pounds per square foot and the velocities in feet per second at the two sections v_m and v_n of a stream flowing through such a closed horizontal channel is given by

$$\frac{p_m}{\delta}+\frac{v_m^2}{2g}=\frac{p_n}{\delta}+\frac{v_n^2}{2g}, \quad \ldots \ldots \quad (42)$$

where δ is the density of the water in pounds per cubic foot. The volume of water flowing through any section is in cubic feet per second, if a represents the area of a section in square feet,

$$a_m v_m = a_n v_n = a_n \sqrt{\frac{2g(p_m - p_n)}{\delta\left(1-\frac{a_n^2}{a_m^2}\right)}}. \quad \ldots \quad (43)$$

[1] See Jamieson's *Applied Mechanics*, Vol. II., page 458.

With suitable manometers or with gages the pressures p_m and p_n can be obtained, and since all other quantities can be represented by a constant, **k,** we have

$$\textbf{Volume per Unit of Time} = k(p_m - p_n). \quad . \quad . \quad (44)$$

The exceptional accuracy of this instrument for measuring the flow of water is well illustrated by the curve in **Fig. 149,**

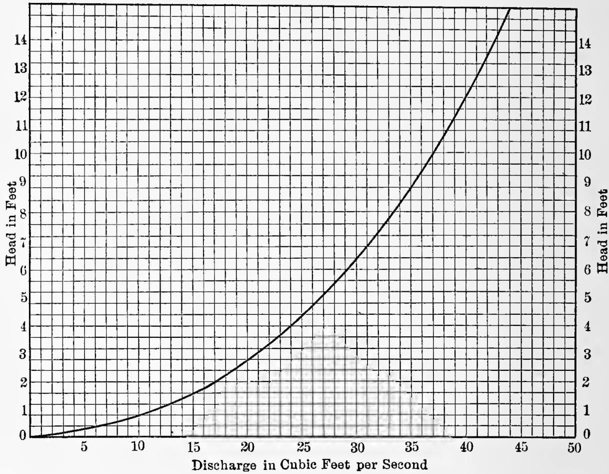


FIG. 149.—Typical Curve Showing Extreme Accuracy of Venturi Meter.

showing the flow as **calculated** from careful measurements of the head, while the barely perceptible dots shown on the curve indicate the results of actual observations by Pullen with a venturi meter.

Flow of Water through Orifices and Nozzles. Theoretically the velocity of flowing water under any pressure is the same as the velocity attained by a body falling freely through a distance equal to that head (**h**) as in **Fig. 150.** Furthermore this statement would be the same even if the water had no free surface, provided, however, the **pressure** at the orifice was that due to a head **h.** If then there is no loss of head due to friction

and eddies formed by the water passing through the orifice, the velocity of discharge, **v**, in feet per second is,

$$v=\sqrt{2gh}, \qquad (45)$$

where **g**[1] is the acceleration due to gravity and **h** is the head in feet.

If **a** is the area of the cross-section of the orifice in **square feet, q is** the quantity or volume of water discharged in **cubic**

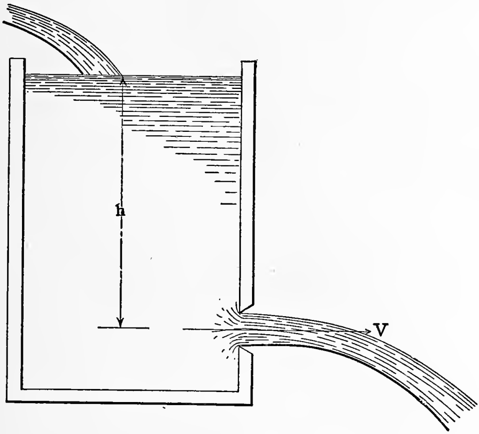


FIG. 150.—Discharge of Water from an Orifice.

feet per second, and assuming the stream is of the same cross-sectional area as the orifice, then,

$$q=a\sqrt{2gh}. \qquad (46)$$

Since the actual flow is less than the theoretical in most cases, and considerably less when the discharge is from a hole with sharp edges in a thin plate, a more general form may be written by inserting a suitable coefficient[2] of discharge, **k,** then,

$$q=ka\sqrt{2gh}. \qquad (47)$$

[1] The value of **g** is approximately 32.2, so that equation (45) can be simplified into $v=8.02\sqrt{h}$.

[2] This coefficient is often called the coefficient of contraction.

Calibration of Orifices and Nozzles. Water under a constant pressure is often measured by observations of the flow through either orifices or short nozzles which have been carefully calibrated. The apparatus required consists usually of a suitably arranged standpipe to which the orifice or nozzle can be attached so that a given head of water can be maintained [1] and a tank on scales (or calibrated for volumes) to receive the water discharged.

A pressure of one pound per square inch is equivalent to a head of water at 62 degrees Fahrenheit of 27.72 inches, or 2.31 feet. A **normal atmospheric pressure** (14.7 pounds per square inch) is therefore equivalent to a **head of 33.96 feet of water.** Then for a given pressure or head the quantity of water discharged in a given time is readily obtained and the coefficient of discharge can be computed by substituting the values of quantity of discharge **q,** the head **h,** and the area **a,** in formula (47).

Data and results should be tabulated in the form given below. The relative roughness of the edge of the orifice or of the inside surface of the nozzle should be recorded.

FLOW OF WATER

Form........................
Flow of water through a...
Date...........................Observers.......................
Form of orifice or nozzle......Formula.......Diameter, feet..........
(Sketch) .... Area, square feet....

No. of Reading.	Head in Feet.	Time in Seconds.	Total Pounds or Cu. feet.	Pounds per Second.	Cu. feet per Second.	Coefficient of Discharge (k)	Remarks.
Average							

[1] In many places a suitable pressure tank is not available, and in such cases the calibration can be made by attaching the orifices or nozzles to pipes carrying water under pressure. The readings of the pressure gage can be reduced to the equivalent head in feet, to which must be added, if the centre of the gage is higher than the orifice or nozzle, the distance in feet from the center of the nozzle to the center of the gage.

Curves. Curves should be plotted for each orifice or nozzle with head for abscissas and (1) the discharge (cubic feet per second) and (2) the coefficient of discharge for ordinates.

Flow of Water over Weirs. When large quantities of water are to be measured, then orifices are unsuitable and it is customary to pass the whole body of water over a **weir** or **gage notch.** This consists of a board placed across the stream so that all the water must pass over it. The length of the notch is usually made less than the width of the stream to give definite conditions. This is accomplished most easily by sawing the notch out of a long board and beveling the edges.

A typical arrangement for measuring the head of water on a weir is illustrated in **Fig. 151.** The head must be determined with great accuracy, and this is done usually by means of a **hook-gage, Fig. 152,** and a suitable level.

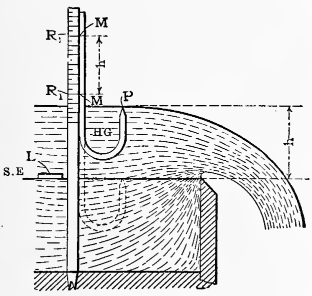


FIG. 151.—A Weir for Measuring Water.

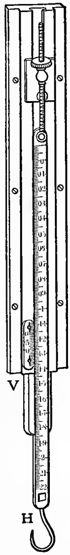


FIG. 152.—A Hook Gage.

The hook-gage consists of a sharp-pointed hook **H,** attached to a vernier scale **V,** intended to measure very accurately the amount the hook is moved. Before taking an observation the hook must first be submerged and then raised slowly till the point just breaks the surface of the water. The correct height of the surface is obtained at the instant when the point of the hook pierces it. The head **h** of the water flowing over the weir (**Fig. 151**) is obtained by setting by means of a straight-edge

SE and the level **L** the point of the hook at the same level as the crest of the weir. The height observed in this position is called the **zero head.** It is to be subtracted from all other readings to get the head of water flowing. The hook-gage must be placed in such a position on the upstream side of the weir where the surface has no appreciable velocity and where there is very little disturbance due to eddies. In terms of the following symbols,

q = quantity or volume of water discharged in cubic feet per second;
h = the head in feet on weir measured in still water;
b = breadth of the weir in feet;
n = the number of contractions;
k = coefficient of discharge.

$$q = 2/3 k h^{3/2} (b - 0.1 n h) \sqrt{2g}. \quad \ldots \ldots \quad (48)$$

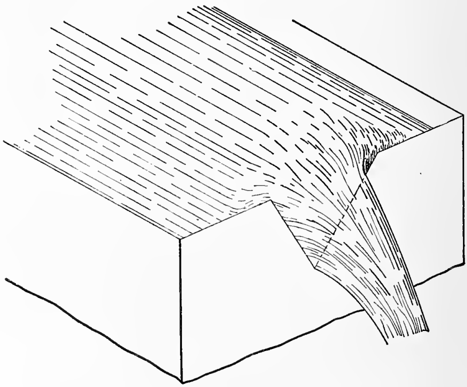


Fig. 153.—Weir with a Triangular Notch.

This is the well-known Francis formula for a rectangular notch. The ordinary rectangular notch has two contractions, one at each end. Triangular notches in weirs are sometimes used. One of these in the form of a right-angled isosceles triangle is shown in **Fig. 153.** It has the advantage of giving the same form of stream whatever the size of the notch or the height of

water passing through. It is, therefore, particularly suitable for measuring a flow of water which is somewhat variable. The quantity of water discharged over a triangular weir or notch is,

$$q = 4/15 kbh^{3/2}\sqrt{2g}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (49)$$

When the angle is 90°,

$$b = 2h \quad \text{and} \quad q = 4.26\ kh^{5/2}. \quad . \quad . \quad . \quad . \quad . \quad (50)$$

Also when the angle is 60°,

$$b = 2h \tan 30° \quad \text{and} \quad q = 2.47 kh^{5/2}. \quad . \quad . \quad . \quad . \quad (51)$$

Any mistake made in determining **h** will produce a larger percentage error in the results with the rectangular and triangular notches than with an orifice. Where great accuracy is desired and the quantity of water to be handled is not too large, an orifice calibrated and used in the bottom of the tank as shown in **Fig. 154** is to be preferred to measurements with a weir. This remark is particularly applicable in connection with the measurements of cooling (circulating) water in tests of large steam engines and turbines.

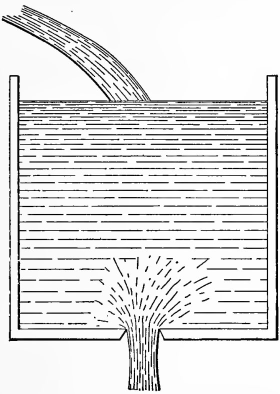


Fig. 154.—Best Kind of Orifice for Engine Tests.

Use the same form for data as given for Calibration of Orifices or Nozzles on page 158.

Curves should be plotted with heads for abscissas and (1) the discharge (cubic feet per second) and (2) the coefficient of discharge for ordinates.

CHAPTER VIII

CALORIFIC VALUE OF FUELS—SOLID, LIQUID, AND GAS

Calorific Power is a term applied to the quantity of heat generated by the complete combustion of a definite quantity of fuel. In order to insure rapid and complete combustion the fuel is **preferably** burned in an atmosphere of oxygen under pressure. This calorific power of fuels is expressed in the English system as **British thermal units per pound,** and in the metric system as **calories per kilogram.**

The quantity of heat generated by combustion is measured by the rise in temperature of a given weight of water in a calorimeter, of which the cooling effect or **water equivalent k** has been determined, and the temperature of the gas escaping has been reduced to that of the room. Now if,

w_f = weight of the fuel in pounds,
w_w = weight of the water in pounds,
k = water equivalent [1] of the calorimeter, in pounds,
t_1 = initial temperature of water, degrees Fahrenheit,
t_2 = final temperature of the water, degrees Fahrenheit,
Q = heat generated, B.T.U.,

[1] Water equivalent is used to express the heat-absorbing effect of the calorimeter as equivalent to that of a weight of water. This may be found as for calorimeters used for determining the quality of steam by the hot-water method (see page 64), by taking the sum of the products of the weights and specific heats of the various parts of the calorimeter (see *Calorific Power of Fuels*, by H. Poole, pages 14 and 15), or by comparing the results obtained with those that should have been secured, if there had been no absorption of heat, by the combustion of some fuel of which the heat value is known; as, for example pure carbon in oxygen gas.

Corrections for radiation can be practically eliminated by having the temperature of the water in the calorimeter before ingition as much below the "room" temperature as the final temperature is above.

then the calorific value **H** per pound of fuel in British Thermal Units is,

$$H = \frac{Q}{w_f} = \frac{(w_w + k)(t_2 - t_1)}{w_f}. \quad . \quad . \quad . \quad . \quad . \quad . \qquad (52)$$

Bomb Calorimeters. Formerly the calorimeters used for burning fuels in an atmosphere of oxygen were arranged for combustion at **constant pressure,** but since it was found that more reliable results could be obtained generally with apparatus maintaining a **constant volume,** the former type is not now much used. When the combustion takes place at **constant volume,** the vessel receiving the charge of fuel and oxygen must be designed to withstand a great pressure, and therefore on account of the massive construction required the vessel is called a **bomb calorimeter.** The essential part of such a calorimeter is the strong steel vessel or bomb similar to **Fig. 155.** It consists essentially of a steel shell **S** capable of resisting with safety a pressure of about 750 pounds. This shell is usually provided with a coat of enamel or a lining of nickel on the inside and is nickel-plated on the outside. The coating or lining on the inside is intended to resist corrosion and oxidizing action during the combustion. The advantage of the nickel lining over the coat of enamel is that when it is worn out or broken it can readily be replaced and at much less expense than the enamel. The shell is closed at the top by an iron cover or cap which is to be made tight by screwing down on a lead washer with considerable force, using a long wrench. At the top of this cover or cap there is a conical seated valve, which is screwed in through the gland and stuffing-box **G,** by attaching a wrench at **P.** The valve and its seat are made of good nickel, as this metal is not easily oxidized. A wire electrode, which is well insulated from the cover, extends into the shell and conducts the electric current for firing the charge of fuel, which is placed on a platinum dish

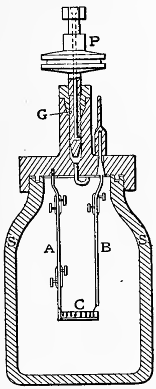


FIG. 155.—Section of a Bomb Calorimeter.

or crucible supported by another wire attached to the cover on the inside.

Usually one gram of fuel is put into the dish to make a test for calorific value. A small iron wire (which was previously weighed) is then suspended over the dish between the electrode and the wire support for the dish. The cover should then be screwed on with a long wrench, the shell itself being held in a vise. The **complete Mahler apparatus** is shown in **Fig. 156,** showing the cylinder of oxygen **O,** the pressure gage **M,** the calorimeter vessel **D.** The end of the conical-seated valve (**Fig. 155**) is attached by means of pipe connections, preferably

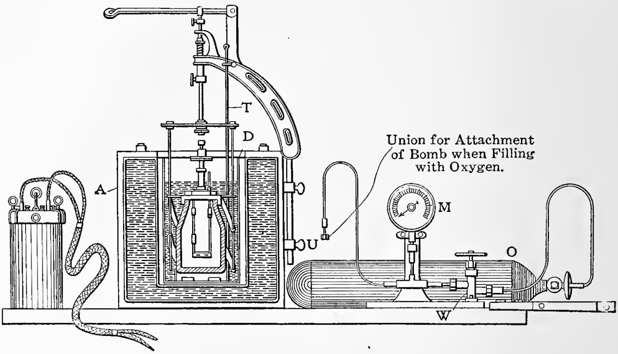


Fig. 156.—Complete Mahler Apparatus.

flexible, to the union **U** and to the valve **W,** which because of the high pressure should be opened slowly and carefully, and allow sufficient oxygen to pass into the bomb to provide a considerable excess above that actually required. The pipes for connecting the bomb to the oxygen cylinder should connect also with a pressure gage as shown, so that the pressure in the bomb can be regulated. For the combustion of coal a sufficient volume of oxygen is admitted to Mahler bombs of the usual size to make the pressure in the bomb from 200 to 300 pounds per square inch. Now close the valve on the oxygen cylinder and the conical seated valve on the bomb, removing also the connections between the bomb and the oxygen cylinder.

The fuel, especially if it is coal, should not be too fine, because if accidentally the oxygen should be allowed to go in a little too rapidly, some of sample of the coal will be blown out of the dish and will probably not be burned. The bomb should then be placed in the calorimeter vessel **D,** which should then be filled with a quantity of water previously weighed to fill it to about the level indicated in the figure. Place the calorimeter thermometer **T** into the vessel, being careful that the end will not be touched and broken by the stirrer or other parts, and then after agitating the water for a few minutes to establish a uniform temperature, the observations can begin. The temperature should be very carefully observed for five minutes and recorded **minute by minute,** to determine the rate of variation of temperature before combustion. Then the electric circuit should be made and the combustion will, of course, begin immediately; but some little time will be required for the transmission of the heat generated to this water. Now take the temperature at the end of a minute after making the electric circuit; and continue observing the temperature every minute till it reaches its maximum value and begins to fall off regularly. Continue the observations for five minutes more to determine the rate of the fall of the temperature. The stirrer should be worked continuously but not too rapidly throughout the test, being careful, however, that the thermometer is not broken. When the observations have been finished, the conical-seated valve should be opened first to relieve the pressure and then the cover or cap can be unscrewed and removed.[1]

The method described for the use of the Mahler bomb calorimeter can be applied also for determinations of calorific value of liquid fuels. Heavy oils can be weighed directly in the platinum dish or crucible, but light oils which are easily vaporized must be put into specially prepared glass bulbs which are broken to allow access of the oxygen, just before the cover is

[1] Some engineers wash out the inside of the bomb with a little distilled water to collect the nitric and sulphuric acids formed. Usually, however, this correction for acids is not made, as the heat liberated in the formation of the acids is usually less than one-third of one per cent, which, of course, would be subtracted from the calorific value obtained. If the reader is interested he will find the method explained with the necessary data in the *Calorific Power of Fuels*, by H. Poole, page 62.

put on the bomb. If sufficient oxgyen is provided in every case there will be complete combustion in the calorimeter with no other refuse than the cinders remaining.

A specimen calculation is given below:

Weights,—coal, .0030 lb.;[1] water in calorimeter, 4.85 lbs.; water equivalent of bomb, etc., 1.10 lbs. Weight of iron wire, .0002 lb.

Preliminary Observations.

Beginning	60.23° F.,	3 minutes, 60.26° F.,
1 minute,	60.24° F.,	4 minutes, 60.27° F.,
2 minutes,	60.25° F.,	5 minutes, 60.28° F.

Rate of variation before combustion, $a_0 = \frac{60.28 - 60.23}{5} = .01°$ F.

Observations during Combustion.

6 minutes, 65.45° F.
7 minutes, 67.29° F.,
8 minutes, 67.38° F., max.[2]

Observations after Maximum was reached.

9 minutes, 67.34° F.,	12 minutes, 67.28° F.,
10 minutes, 67.32° F.,	13 minutes, 67.27° F.,
11 minutes, 67.30° F.	

Rate of variation after maximum, $a_m = \frac{67.38 - 67.27}{5} = .022°$ F.

The rate of variation of temperature before combustion was for cooling the water and that after combustion was for a loss of temperature by the water. Evidently, then, the two rates are opposed in effect and the true **average rate of variation** is

$$a_v = \frac{-0.1 + .022}{2} = +.006° \text{ F per minute.}$$

[1] The coal had been warmed at a temperature of about 240 to 280 degrees Fahrenheit before weighing, in a crucible over a Bunsen burner or an alcohol lamp to drive off the moisture.

[2] Some engineers make a curve of temperatures (ordinates) and time (abscissas) and use for the final temperatures in the calculations the value from the curve, when the part of the curve representing the cooling becomes a straight line. The difference in numerical values is usually very slight.

Three minutes (5–6, 6–7, and 7–8) were required for complete combustion or for the water to reach the maximum temperature. Total cooling correction to be added to the observed rise in temperature is therefore, $3 \times .006 = .018°$ F.

The total rise as corrected is, $7.10 + .018 = 7.118°$ F.

The quantity of heat generated is, therefore, $Q = (4.85 + 1.10) \times 7.118 = 42.35$ (B.T.U.) for .0030 lb. of coal; and from this result must be subtracted the heat of combustion of the iron wire $.0002 \times 3000$ [1] or 0.60 B.T.U. The net value of the heat generated from the coal is, therefore, $42.35 - .60 = 41.75$ B.T.U.

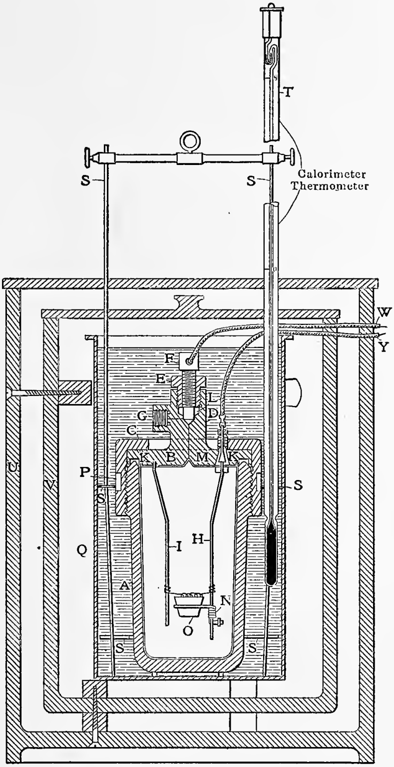


Fig. 157.—Atwater's Fuel Calorimeter.

A modification of the Mahler bomb calorimeter has been designed by Atwater,[2] **Fig. 157,** and another by Emerson, **Fig. 158.** The **former** consists of the shell of the bomb **A,** the cap **C,** screwed on numerous threads to the shell, and holding down the cover **B.** Into the vertical neck of this cover a screw **E,**

[1] The calorific value of pure iron is about 3000 B.T.U. per pound.
[2] Atwater, *Bulletin No.* 21, U. S. Dept. of Agriculture.

holding another screw **F,** is fitted and is to be turned down tightly, a lead washer serving as "packing." A small passage for the admission of oxygen from **G** is opened and closed as required by turning the screw **F** operating a needle-valve. A wire **H** of platinum or other non-oxidizable metal passes through the cover **B** and is insulated from it by a collar of hard rubber. Another wire rod **I** is attached to the lower side of the cover and electrical connection is made between the two wires **H** and **I** by a small iron wire stretched between them. A platinum crucible provided for receiving the fuel is supported by a "screw" ring. Ball bearings of hard steel are sometimes placed between the cover and the cap to reduce friction when screwing down. Holes located in the sides of the cap are for the attachment of a long spanner wrench when turning down the cap. A hand-stirring device **S** is used for agitating the water in the vessel **Q.**

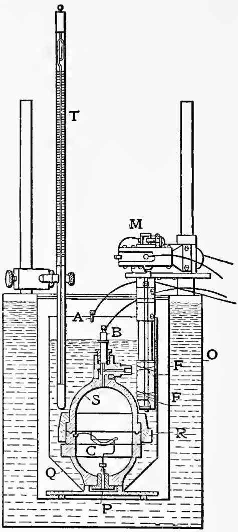


FIG. 158.—Emerson's Fuel Calorimeter.

The usual arrangement of the oxygen tank, pressure gage and tubing for charging an Atwater calorimeter is illustrated in **Fig. 159.** A pellet press for compressing samples of fuel into a suitable size to burn in the crucible of this calorimeter is shown in **Fig. 160.**

Fig. 158 shows another form of bomb calorimeter (Emerson) of which the Mahler is typical. It consists of a nearly spherical shell **S,** divided into two parts which are screwed together by the ring **R.** Powdered fuel is placed in the crucible **C** and is ignited electrically by the current passing through the water in the vessel **Q** from the terminal at **A,** then through an insulated contact point **P** in the bottom of the calorimeter to a small platinum or iron wire in the crucible **C,** which becomes heated by the passage of the current to a white heat, igniting

the fuel. One end of this small wire is fastened to make electrical contact with the lining of the calorimeter, which in turn is connected electrically with the plug and terminal at **B.**

The outer vessel **O** is to be filled with water to the top. The stirring device consists of small propellers **F**, on a vertical shaft operated by a small electric motor **M.**

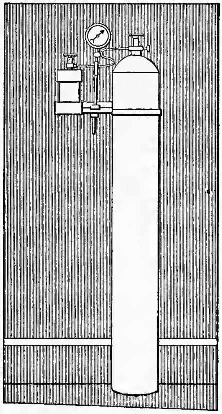


FIG. 159. — Apparatus for Charging Atwater's Calorimeter with Oxygen.

Parr Calorimeter. It is not always convenient to secure a supply of oxygen under pressure for use in a Mahler bomb, and consequently another type of fuel calorimeter, known as Parr's, has found considerable use, especially for relative determinations in power plants. The results obtained can never be depended on to be as nearly accurate as determinations with one of the bomb type. **Fig. 161** illustrates a simple form of Parr calorimeter. Sectional views of the two kinds of calorimeter vessels used are shown in **Figs. 162** and **163.** In the former the ignition is accomplished by dropping a hot wire through the neck into the shell **A** of the calorimeter. The cover is attached to the shell by means of a threaded nut **F.** A charge for the bomb consists of about .004 pound of pulverized coal from which the moisture has been driven off by warming for about an hour at a temperature of about 240 to 280 degrees Fahrenheit, and **eighteen** times as much by weight of sodium peroxide, which supplies the oxygen needed for com-

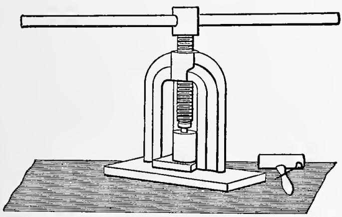


FIG. 160.—A Pellet Press for Compressing Samples of Fuel.

bustion. The charge should be well mixed by shaking the shell after the cover has been securely fastened. The cover must be attached very securely by turning up the nut with a long wrench, while the shell is held in a vise or in some similar manner, because

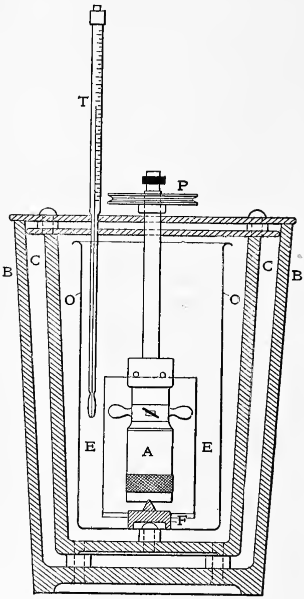


Fig. 161.—Typical Parr Calorimeter.

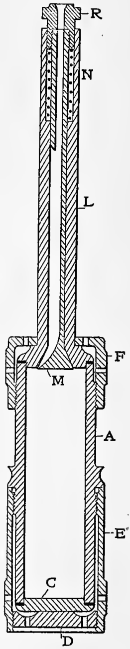


Fig. 162.—Parr Bomb for Hot Tube Ignition.

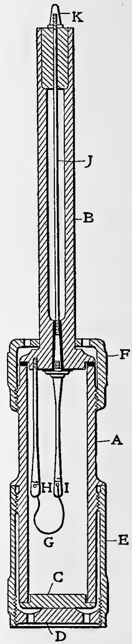


Fig. 163.—Parr Bomb for Electrical Ignition.

there is a violent explosion when ignition takes place. When the hot wire is put into the tube in the long neck **L,** the cap **R** at the top must be struck quickly with a mallet before the wire cools in order to open the valve **M,** which opens inward into the shell and permits the wire to fall through. To be certain of obtaining a good result the wire should be heated almost

to a white heat. The rise of the mercury in the thermometer will indicate when an explosion has occurred.

The calorimeter is provided usually with wings or small propeller blades for agitating the water in the vessel **O.** The small pulley **P** (**Fig. 161**) shown at the top of the neck is used for turning the calorimeter bodily in the water when supported on

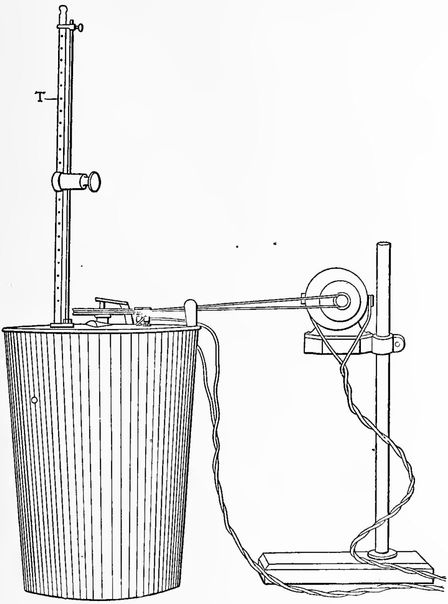


FIG. 164.—Parr Calorimeter with Motor Stirring Device.

the pivot **F,** shown at the bottom of the figure. The **water equivalent** of the calorimeter is determined in the same way as for other fuel calorimeters. **Fig. 164** shows the Parr calorimeter as designed for electrical ignition and with a stirring device operated by an electric motor.

Allowance must be made in the calculations for the heat of combustion of the sodium peroxide, which for the proportions given is approximately 27 per cent of the heat generated.

Carpenter's Calorimeter. There are few calorimeters in which the oxygen is supplied at constant pressure which are altogether successful. One of the best forms of an apparatus of this kind has been designed by Carpenter, especially for coal determinations. With this apparatus no thermometers are needed, as the rise in temperature is measured by the expansion of the mass of water surrounding the combustion chamber.

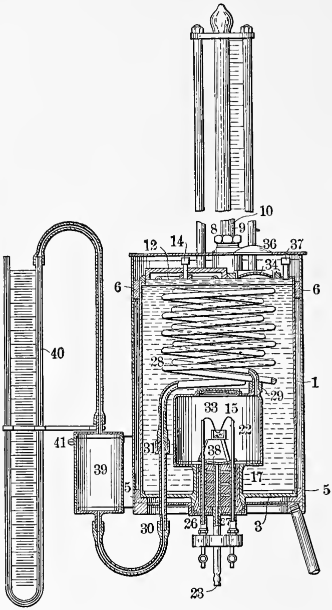


FIG. 165.—Carpenter's Calorimeter.

This apparatus is shown in **Fig. 165.** It consists of a combustion chamber **15,** provided with a removable bottom **17,** through which the tube **23,** supplying the oxygen, passes into the combustion chamber. Electric current for ignition is conducted through the wires **26** and **27.** The removable bottom supports also the asbestos cups or crucibles **22,** used for holding the sample of coal to be burned. Just beneath the crucibles a silver mirror **38** is provided to deflect the heat. The plug containing the wires and the oxygen pipe **23** is made of alternate layers of asbestos and vulcanite. Products of combustion leave the combustion chamber through a spiral tube, the parts of which are marked **28, 29, 30,** and **31,** into the small vessel **39,** attached to the outer casing of the instrument, and are finally discharged into the air from a small hole **41,** in the side of the vessel. The pressure in the chamber **39** is indicated by a manometer gage **40.** The inner casing of the instrument **1,**

containing the water for absorbing the heat generated, is nickel-plated and highly polished to reduce radiation as much as possible. An open glass water gage **10** passes through the casings and extends below the water level. This water gage, with the scale attached to it, replaces the thermometer used in other calorimeters for measuring the rise in temperature. The scale is graduated to read inches, and it is calibrated usually by burning coke in the calorimeter, and determining thus the rise of the water level for a determinable weight of pure carbon. A calibration curve is usually supplied with the instrument. By moving the diaphragm **12**, by means of the screw **14**, the water level can be regulated, as well as the "zero" level in the glass water gage **10**. A funnel **37** is provided for filling the instrument, and by inverting it, this funnel can be used also for draining. The instrument holds 5 pounds of water, and 2 grams of coal is the amount taken usually for a charge, requiring about twenty minutes for complete combustion; powdered coal is used. The asbestos cup should be heated in the flame of a Bunsen burner before it is weighed. The charge of dried coal should then be put into it and weighed again. The difference will be the weight of the coal used. Now put the charge into the combustion chamber **15**, place the platinum ignition wire above the coal, connect wires **26** and **27** to the battery, and as soon as the heat generated causes the level of the water to rise in the glass water gage **10**, open the valve in the pipe discharging oxygen into tube **23**, and then by pulling down the platinum wires to touch the contents of the crucible, the coal will be kindled. At the same time the reading of the glass scale opposite the gage glass **10** must be observed and recorded. Progress of the combustion can be observed through the glasses **33**, **34**, and **36**, arranged vertically over each other for this purpose. As soon as the combustion is complete observe the time and the reading of the scale opposite the glass water gage **10**. The difference between this last reading and the one taken at the beginning of the test is called the "actual" scale reading.

The correction for radiation is made by observing the reading of the scale of the water gage after the oxygen has been shut off, a length of time equal to that required for the combustion. The difference between this reading and the "actual" reading

is to be added to the "actual" reading to obtain the corrected reading.

By weighing the asbestos cup after the test is finished and subtracting from this weight that obtained previously for its weight empty, the weight of ash is determined.

In order that Carpenter's calorimeter may give determinations of heat values that are at all accurate, all the air must be removed from the water used, as the presence of air[1] will affect the relative level of the water in the gage glass for a given rise in temperature. The oxygen must also be supplied at a constant pressure, maintaining the pressure indicated by the manometer gage at the value for which the calorimeter was calibrated. Most calibration curves are made for a pressure of about 10 inches of water. The apparatus can be made to give good comparative results when operated carefully and "according to directions." In general, the statement is often made that coal calorimeters intended for combustion at constant pressure will usually give nothing more than "faint approximations" to correct results. The same can be said, however, of nearly all the calorimeters if they are not very carefully manipulated.

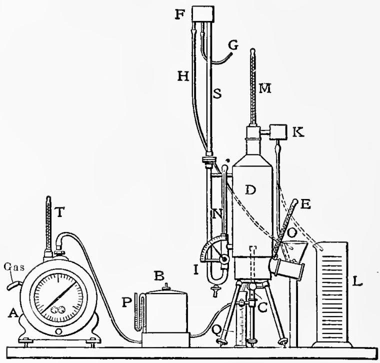


FIG. 166.—Junkers Calorimeter with Auxiliary Apparatus.

When making calorific determinations of coal the distinction must be carefully made between results obtained per unit weight of combustible or per unit weight of coal (including moisture and ash).

Junkers Calorimeter for Liquids and Gases. An apparatus for determining the calorific power of gases is shown in **Fig. 166** and **Fig. 167.**

[1] About 2 inches of kerosene oil is usually put into the glass water gage to prevent air from coming into contact with the water.

The gas flowing in pipes at the left (**Fig. 166**) passes through the meter **A,** then through the regulator **B,** and is burned in a type of Bunsen burner **C,** in the lower part of the calorimeter. This instrument consists of a cylindrical copper vessel through which water is constantly circulating. The gases from the Bunsen flame in the calorimeter pass up through the hollow central portion of the instrument and near the top are deflected downward through a group of small tubes arranged in an annular ring between the outside and inside walls of the calorimeter. Around these tubes water is kept circulating continuously to absorb the heat generated by burning the gas tested. After leaving these tubes the products of combustion discharge first into a chamber **31** (**Fig. 167**) and then into the air through the flue **D.** In order to keep the flow of water as regular as possible it is brought from the supply pipe **G** into a small reservoir in which the water is kept at a constant level (constant head) by means of an overflow pipe **H.** The water supplied to the calorimeter passes down through the pipe **6,** through a valve at **I,** and discharges at **K,** running into a vessel in which it is weighed. A graduated tube **Q** (**Fig. 166**), is provided to collect the moisture from the steam that is condensed. The condensed steam collects in the combustion chamber **31** and escapes through the tube **35.** A thermometer **N,** in a cup near the valve **I,** indicates the temperature of the water entering the calorimeter, and one at **M** shows the temperature of the water leaving. The temperature of the products of combustion (burned gases) is indicated by the thermometer **O,** in the gas flue. The calo-

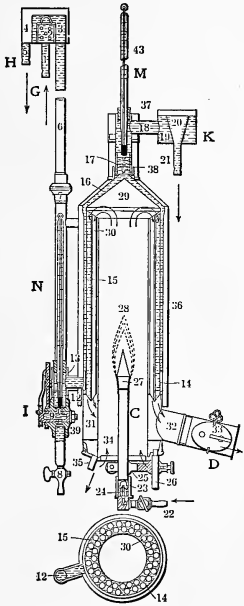


Fig. 167.—Section of Junkers Calorimeter.

rimeter is provided with an air jacket and is covered with sheets of copper, nickel plated and highly polished so that the radiation loss is considered negligible. If, then, the flow of water and the rate of burning the gas are regulated so that the **temperature of the products of combustion**[1] **as indicated by the thermometer at O, is the same as the temperature of the air surrounding the calorimeter,** practically all the heat generated by the burning gas is absorbed by the water. The rise in temperature of the water is observed by reading the thermometers at **N** and **M.**

Now if the temperatures of the water at the inlet and the discharge have been observed and the weight of the water flowing has been determined while, for example, a cubic foot of gas[2] has been burned, then the difference in temperature in degrees Fahrenheit times the weight of water in pounds gives the heat value in British thermal units per cubic foot of gas.

For some calculations relating to the efficiency of heat engines it is desirable to know the number of heat units representing the calorific value of the gas when the steam formed in the combustion is not condensed but is carried off with the products of combustion. To determine this value, sometimes called the **"low heat value"** of the gas, the latent heat at atmospheric pressure of the amount of condensed steam collected must be subtracted from the value obtained by multiplying together the rise in temperature and the weight of water used. This correction is usually about two per cent. This apparatus, although it operates by a constant pressure method, gives very satisfactory determinations.

Fig. 168 shows a balance and lamp attachments for a Junkers calorimeter set up for determining the heat value of liquid fuels like gasoline, kerosene, crude oil, etc. The heat

[1] Heat lost in products of combustion is explained in Stillman's "Engineering Chemistry," pages 161–165. See also in this book pages 196 and 224–226.

[2] In order that results can be compared, it is customary to reduce the calorific power to terms of heat units per cubic foot of gas at a "standard" temperature and pressure as for example, 32 degrees Fahrenheit and 14.7 pounds per square inch, assuming that the volume of the gas is proportional directly to the temperature and inversely to the pressure, or else by determining the specific volume of the gas to calculate the heat units per pound.

generated is measured in the same way as when gas is burned and the weight of oil used is determined by weighing on the balance to which the lamp **L** is attached.

Calorific Values from Chemical Analysis. Dulong stated a long time ago that the heat generated by burning any fuel was equal to the sum of the "possible heats" generated by its component elements, less that portion of the hydrogen which combined with the oxygen in the fuel to form water. When hydrogen and oxygen exist together in a compound in the proper proportions to form water, the combination of these elements has no effect on the calorific value of the compound. Now the calorific value of a pound of carbon is 14,600 B.T.U. and of a pound of hydrogen is 62,000 B.T.U., so that by Dulong's formula, the calorific value of a pound of fuel would be stated, using these values, as

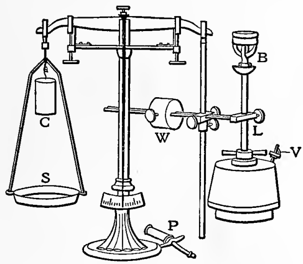


FIG. 168.—Balance and Lamp for Burning Oils in Junkers Calorimeter.

$$x = 14{,}600C + 62{,}000\left(H - \frac{O}{8}\right) + 4{,}000\,S,$$

here **C, H, O** and **S** are respectively the weight of the carbon, hydrogen, oxygen and sulphur in a pound of fuel. As the result of testing the forty-four different kinds of coal with his bomb calorimeter Mahler developed the following formula, using the same symbols used in Dulong's,

$$x = 200.5 + 675H - 5{,}400.$$

Using this latter formula Lord and Haas[1] determined for a series of 40 Pennsylvania and Ohio coals which they had analyzed and computed the calorific values that the maximum differences between the calculated results and the determinations with a bomb calorimeter were from 2.0 to − 1.8 per cent. With fuels like coke, charcoal, and anthracite coal, in which the

[1] Trans. *American Inst. of Mining Engineers*, Feb., 1897.

content of volatile matter is small, the calorific values calculated from an accurate analysis are usually in very close agreement with accurate calorimeter tests, but with coals having more than 20 per cent of volatile matter there is likely to be considerable error.

Proximate Analysis of Coal. For all tests in which an analysis of the coal or its calorific value is to be determined, it is very necessary that the sample to be tested be selected with the greatest care. The method generally adopted for obtaining a fair sample is known as "quartering," as explained in the **Rules for Conducting Boiler Trials** adopted by the American Society of Mechanical Engineers. (See page 210.) The utmost care must be taken that the amount of moisture in the sample received for analysis is the same as that in the original condition, or more specifically in a **boiler test,** at the time when the coal used in the test was weighed. For this reason samples of coal should be transported and stored in air-tight preserving jars or similar vessels. It is not unusual, moreover, to find that coal containing 10 per cent of moisture will lose as much as 2 or 3 per cent of its moisture in the process of careless sampling, crushing, resampling, etc., while if it is allowed to remain exposed to atmospheric conditions for a considerable time in a warm room as much more may be lost by evaporation.

Moisture Determinations. Determinations of moisture are made by careful engineers as soon as permissible after the sample has been procured and with the coal in as large pieces as possible.

A good method for making the moisture determinations for **anthracite** and **semi-bituminous** coals is to place a weighed sample on top of the hottest part of a boiler setting or a flue and weigh it again after drying for twelve hours. A good laboratory test for the same kinds of coal is to place a sample weighing about .05 pound in an air or sand bath for one hour at a temperature of from 220 to 230 degrees Fahrenheit, and weigh again when the sample is cool.[1] The difference in weight is the amount of moisture in the sample.

[1] While the sample is cooling there is the possibility that it may absorb moisture from the air unless it is placed in a desiccator till cool. It is difficult to get accurately the weight of hot bodies on account of the air currents produced.

When coals taken from the mines west of the Pittsburg district or other coals containing inherent moisture are to be tested for moisture, a different method must be adopted. The sample of coal is to be spread out in a thin layer and exposed for about four hours to the atmosphere of a warm room. The difference between the weighings before and after this exposure is the weight of surface moisture. Then crush all of this sample to produce coarse grains measuring not more than one-sixteenth inch on a side, mix it thoroughly, and select from it a quantity weighing from .05 to .125 pound, and dry it for one hour, in an air or sand bath in which the temperature is maintained at from 240 to 280 degrees Fahrenheit. Now weigh it again and then continue the heating between these limits, weighing every hour till two of these weighings are the same or the weight begins to increase due to oxidation of the coal. The difference between the original and the minimum weight of this sample is called the moisture in air-dried coal. The sum of the percentage of surface moisture plus the percentage of moisture in air-dried coal is the total moisture. When making the determination for moisture too much care cannot be exercised to remove all of it.

Determination for Volatile Matter. The amount of volatile matter in coal is determined usually with a sample as originally received without drying. A suitable sample should weigh about .0035 pound—about 1.5 grams—which should be pulverized in a mortar and put into a clean platinum or porcelain crucible. Then weigh the crucible with the coal it contains in a balance sensitive enough to weigh accurately to one-thousandth pound. After this weighing has been done as carefully as possible, a cover like the ones usually provided for crucibles of this kind is to be put on to cover it tightly. Now heat the crucible for $3\frac{1}{2}$ minutes over a Bunsen burner, keeping the crucible at a bright red heat, and then immediately, without cooling, for $3\frac{1}{2}$ minutes over an air-blast lamp. After cooling weigh, and the difference between the weighings is the sum of the volatile matter and the total moisture.

Determination of Fixed Carbon. The determination of the fixed carbon is made by heating again the sample used for the determination of volatile matter.[1] Now, however, the cover

[1] If the sample tested is a bituminous coal it will be observed that coke has been formed by the removal of the volatile matter.

of the crucible is to be removed and heat is to be applied, preferably with a Bunsen burner, until all of the carbon is burned; that is till the weight becomes constant. If the time available for making the test is limited an air-blast lamp may be used instead of the Bunsen burner. The rate of combustion can be increased by stirring the sample from time to time with a platinum wire. What now remains in the crucible is the ash. The difference between the weight found after the volatile matter had been driven off and the weight of the crucible and the ash is the weight of fixed carbon. Weight of ash should be determined by weighing the crucible again when empty.

If sulphur and phosphorus determinations are required they should be made by an expert chemist.

CHAPTER IX

FLUE GAS ANALYSIS

Flue Gas Analysis. The analysis of flue gases in connection with tests of steam boilers gives a valuable means for determining the relative value of different methods of firing and of different types of furnaces. Errors in the analysis of flue gases are most often due to the inability to secure an average sample of the gases in the different parts of a flue or chimney. The composition is likely to vary considerably even during short intervals, and it is therefore desirable to adopt some method of sampling which will permit collecting the sample slowly for a considerable period.

A very simple and convenient sampling apparatus is shown in **Fig. 169.**

The sample of gas is taken from the flue or chimney through the pipe shown at the top of the figure. This pipe extends well into the flue and has usually a long slot cut into one of the sides so that a better sample of gas can be taken than if it were taken at the end of the pipe. The end of the pipe outside the flue is connected by means of a short rubber tube to the sampling bottle. A valve **V** should be put as near as possible to the end of such pipes, so that they can be closed up when the sampling bottle is removed for analysis. If a valve is not provided or is not closed, the suction in the flue will draw air into the pipe, and when again connecting up the sampling bottle this air must be removed before a true sample can be taken. The sampling bottle is preferably one with a wide neck, closed with a cork through which two glass tubes pass into the bottle, one reaching nearly to the bottom and the other entering only a little beyond the bottom of the cork. The longer tube can be connected to an **aspirator** or **ejector,** (**Fig. 170**) a water-jet exhaust or any similar device producing a steady suction.

Small aspirators or ejectors operating on the principle of an injector with a small stream of water which entrains the gases is very convenient for collecting samples continuously. Water enters through a vertical nozzle, entrains air or gas drawn in through the side openings and the mixture of atomized water and air is discharged with considerable velocity through the forcing tube at the bottom.

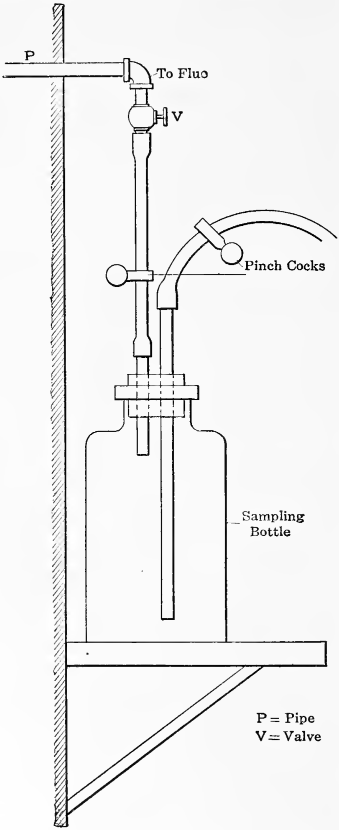


Fig. 169.—Sampling Bottle for Collecting Flue Gas.

If an aspirator is not available, the bottle may be filled with mercury and by making a siphon of the rubber tube attached to the longer of the two tubes in the bottle, the mercury can be gradually drawn out and gases drawn in. By adjusting the valve **V** the rate of flow of the gases into the bottle can be regulated. Mercury is too heavy to use in a very large sampling bottle and therefore water is often used instead, with the disadvantage, however, that the water will probably absorb some of the constituents of the gas. On this account very little water should be left in the bottle, with the sample of gas. If the water is saturated with gases, as it will be from long use, this precaution need not be observed.

The bottle and the tubes must be completely filled with water before beginning to take the sample, because any air left in them will remain in the bottle to be mixed with the sample of the gas. If the end of the cork going into the bottle is made slightly conical it will be easier to avoid entrapping bubbles of air at the top of the bottle.

This type of sampling bottle can be used also very conveniently by reversing the connections of its tubes; that is, by attaching the long tube to the pipe entering the flue, and then turning the bottle upside down. The water will then run out through the shorter tube and the gas will be drawn in to fill the bottle.

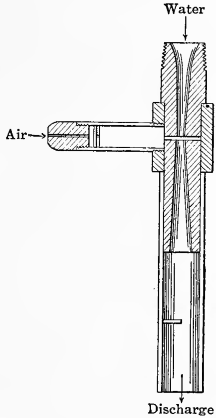


Fig. 170.—Water-jet Aspirator or Ejector.

A portion of the gas can be removed from the sampling bottle into the measuring burette or tube required for making the analysis of the flue gas by connecting the short tube of the sampling bottle to the burette or to some other part of the gas analysis apparatus as may be required. If now the rubber tube connected to the longer tube of the sampling bottle when disconnected is put into a pail well filled with water, then as the gas is withdrawn from the bottle, water will be drawn from the pail to displace it. One of the advantages of this apparatus is that it can be easily made from the materials obtainable in almost any town or village. A two-quart preserving jar with a rubber cork to fit and tubes of glass, brass or iron can be used to make up a very good sampling bottle.

Since there is nearly always a great variation in the composition of the gases in the various parts of a flue or chimney it is not very likely that a tube open at the end and having a long slit like the one described in the preceding paragraphs will give a "fair" sample. Obviously most of the gas will enter the slot in that portion of its length nearest the collecting apparatus. Another device often used for a sampling

tube consists of a horizontal pipe into which a number of branch tubes are fitted. These branch tubes are arranged so that the openings at their ends will take samples from the different parts of the flue or chimney in which they are placed. Sometimes these branch pipes are also slotted or are perforated with small holes drilled into their walls.

Fig. 171 shows an arrangement of sampling tubes for collecting flue gas recommended by the American Society of Mechanical Engineers. It consists of a series of standard one-fourth inch pipes, all open and otherwise alike at the ends and of equal lengths. Each pipe is to be placed with one end in a shallow box or receiver made of galvanized sheet iron. It is convenient usually to make the depth of this sheet-iron box about the same as that of a course of bricks. These tubes should be arranged so that the open ends will be at points well distributed over the area of the flue which in the figure is marked **A.** The other ends, which are also open, are to be enclosed in the receiver **B.** The receiver is connected by four tubes **C, C** with a mixing box **D.** The flue gases drawn from it should be well mixed and should represent an average sample from the flue of chimney from which they are taken. Tests have shown that two such sampling devices placed in the same flue one above the other about a foot apart, will furnish samples of the flue gases showing the same composition when analyzed.

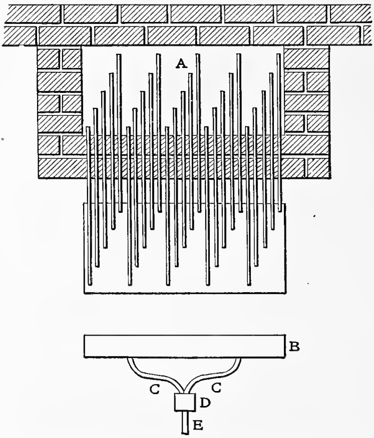


Fig. 171.—A. S. M. E. Arrangement of Sampling Tubes for Flue Gas.

A very convenient type of sampling bottle is shown in

Fig. 172. It consists of a bottle with an opening at the bottom (tubulated), and is provided with a cork at the mouth through which a glass funnel **F** and a tube are passed. The bottle contains water and light oil, and when it is filled there will be a layer of about 4 inches of the oil over the water. The tube **O** at the top is to be connected to the sampling tubes in the flue and the sample is taken in by opening the valve in this tube and also the one at the bottom of the bottle. The water drains off at the bottom and is replaced by the sample of gas. The glass funnel is used for pouring water into the bottle and in this way expelling the gas needed for analysis. The gas is thus made to pass out through the same tube through which it is drawn in.

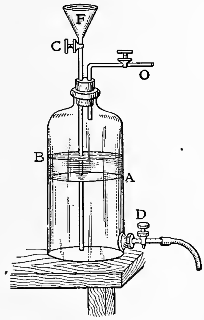


Fig. 172.—Another Type of Sampling Bottle.

Another type of sampling apparatus used by many engineers consists of two galvanized-iron tanks, each about 2 feet high, and about 5 inches in diameter. On the side of each of these tanks and close to the bottom a valve is attached by soldering. These two valves are connected by a piece of heavy rubber tubing. One of the tanks is closed at the top, and a small stopcock or valve is attached to the cover. The other tank is open at the top. The apparatus is used for collecting gas by filling to "overflowing" the tank with the closed top with water from the other tank by raising the latter so that the level of the water in it is above that of the water in the closed tank. By means of rubber tubing the stopcock or valve on the closed tank is then connected to the sampling tubes in the chimney or flues. Meanwhile the open tank is held at such an elevation that the water will not run back into it and create a vacuum in the closed tank. After this connection has been made the stopcock and valves are to be opened again, so that when the open tank is placed below the level of the closed one, the water will flow into the open tank and fill the other one with gas. This operation should be repeated several times before the sample is carried away

to be analyzed, so that there can be no doubt that none of the air in the sampling tubes entered the sample to be analyzed. This water can be used over and over again, and when it has become saturated with gas it is practically as good as mercury for use in collecting the gases.

When a sample of flue gas is taken from the flue at a considerable distance from the furnace it is likely to become mixed with air leaking through the brickwork of the boiler setting, and the analysis will not show the true relations between the volumes of the so-called flue gases and the excess of air. To prevent as much as possible this leakage of air the joints in the masonry must be examined and repaired if necessary and the sample must be taken as near as possible to the fire, bearing in mind, however, that they must be drawn very slowly from the hot flue in order that they will be cooled down gradually to avoid dissociation. For hot gases an earthenware collecting vessel may be used if a glass bottle is likely to be broken. If dissociation occurs in the sample the analysis may show results entirely different from the true composition of the gas in the flue. It is also difficult to prevent the entrance of air into a flue through the bearings of dampers, and whenever it is possible the sample of flue gas should be obtained between the furnace and the damper. At high temperatures sampling tubes of other metals than platinum[1] or nickel are not quite satisfactory, since by their oxidation they abstract the oxygen from the gases passing through them.

Apparatus for the Analysis of Flue Gases. Samples of flue gases contain in varying amounts carbonic dioxide (carbonic acid), oxygen, carbonic oxide, nitrogen, unburned hydrocarbons, and occasionally some free hydrogen. For the data which an engineer usually requires it is not necessary to determine by direct analysis more than three of these; carbonic dioxide, CO_2, oxygen, O_2, and carbonic oxide, CO.

The determination of carbonic oxide, CO, with the facilities and the portable apparatus ordinarily available in engineering laboratories is often somewhat doubtful. Some authorities state that there is rarely more than a trace of carbonic oxide to be found in the gases from combustion in the ordinary

[1] Porcelain and annealed glass are also satisfactory materials to use for making sampling tubes for very hot flues.

types of furnaces. When more than one per cent of carbonic oxide is shown by the analysis and the carbonic dioxide determination is not over 14 per cent, it may usually be assumed that a large part of what is taken to be carbonic oxide is oxygen which was not absorbed by the proper reagent.

In the following table a set of analyses of flue gases is shown. The determinations were made by Scheurer-Kestner with coal from Ronchamp. Other analyses of flue gases may be checked by a comparison with this table. Thus when the analysis shows about 8.2 per cent CO_2, the sum of the percentages of CO_2 and O_2 will probably be between 19 and 20.

PERCENTAGE COMPOSITION OF FLUE GAS

CO_2	O_2	CO	N	Hydrocarbons
8.2	11.3	.2	79.8	.5
10.8	9.0	.2	79.7	.3
12.9	5.5	.2	80.3	1.1
13.4	4.4	.2	80.2	1.8
14.6	2.8	.3	80.6	1.7

In the portable apparatus used by engineers for the analysis of flue gases a separate pipette or treating tube is provided for each reagent, and the chemicals used are of greater strength than the reagents used by some chemists. The following reagents give satisfactory results in a portable apparatus:

(1) For absorbing CO_2 a solution of one part of potassic hydrate (KOH) or caustic potash dissolved in two parts **by weight** of water is generally used.

(2) For absorbing O_2 either an alkaline solution of pyrogallic acid or sticks of phosphorus are employed.

The alkaline solution of pyrogallic acid is prepared by mixing together preferably in the absorption pipette or treating tube, to prevent access of air, 5 grams of pyrogallic acid powder and 100 cubic centimeters of potassic hydrate (KOH) solution prepared as explained above.[1]

(3) For absorbing carbonic oxide a hydrochloric acid solution of cuprous chloride is used. This is prepared by dissolving about 10 grams of cupric oxide in from 100 to 200

[1] When the temperature is lower than about 55 degrees Fahrenheit this reagent does not give satisfactory results.

cubic centimeters of concentrated hydrochloric acid. This solution must be allowed to remain in a bottle tightly closed and well filled with copper wire or gauze, until the cupric chloride is reduced to cuprous chloride. In this latter state the liquid will be colorless. Exposure to the air produces a brown color, indicating the cupric state.

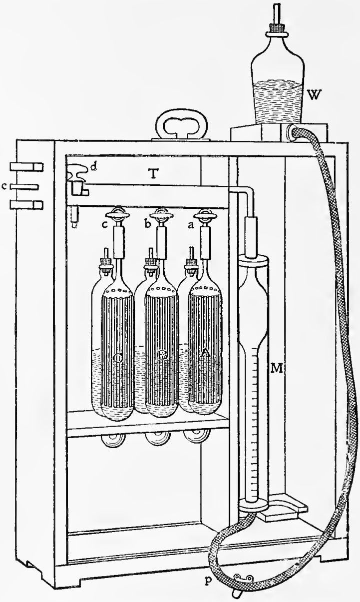


FIG. 173.—Fisher's "Orsat" Apparatus.

After a time these reagents must be replaced by new solutions. The potassium hydrate solution may be used till each volume has absorbed forty volumes of CO_2. Pyrogallic acid solution deteriorates rapidly and each volume should be expected to absorb only one or two volumes of O_2. Cuprous chloride will absorb an equal volume of CO.

Portable devices for the analysis of flue gases are generally known as "Orsat" apparatus. Of these there are various types. The one devised by **Fisher**, shown in **Fig. 173**, has been used extensively. It consists of a measuring-tube **M** surrounded by a water-jacket, and set of absorption pipettes, **A**, **B**, **C**, each filled with a reagent. Each of these pipettes (**Fig. 174**) consists of two glass vessels connected by a U-shaped glass tube at the bottom. One end of these pipettes is joined by means of a short piece of rubber tubing to a glass yoke **T**, which is designed for attachment at one end **e** to a tube leading to the sampling-bottle and at the other end to the measuring-

tube **M**. A water bottle **W** is connected by a flexible rubber tube **P** to the bottom of the measuring tube.

Before a sample of gas is taken into the apparatus for analysis certain adjustments must be made. In the first place, the reagents in the pipettes must all be brought to a standard level at some arbitrary point, usually indicated by a scratch on the glass tube just below the short rubber tube connecting it to the yoke. This adjustment is accomplished by opening one at a time the valves at the tops of the pipette and removing the air (or the gas as the case may be) from it by lowering the water level in the measuring tube. The position of the water bottle determines, of course, the level of the water in the measuring tube. When all the air and gases remaining from a previous test have been expelled from the apparatus by filling the measuring tube and the tubes comprising the yoke with water, one of the tubes in the sampling bottle should be connected to the apparatus at **e** and by opening the valve in the yoke at that end and lowering gradually the level of the water in the measuring tube **M** a sample is obtained for analysis. This sample must be measured by the scale on the measuring tube at atmospheric pressure[1] to the nearest tenth of a cubic centimeter.[2] After the measurement has been made and recorded if the cock in the tube leading to the absorption pipette containing the reagent for absorbing CO_2 is opened and then the water bottle is raised, all of the measured sample of gas can be forced over into

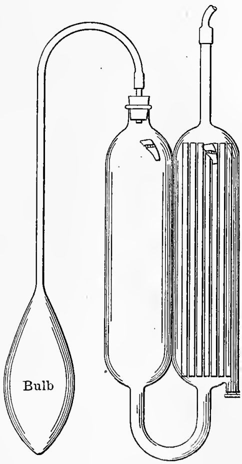


FIG. 174.—Pipette of Fisher's "Orsat" Apparatus.

[1] The pressure of the gas is "atmospheric" when the water bottle is held so that the water in it and that in the measuring tube are at the same level.

[2] The scales of practically all measuring tubes used for gas analysis apparatus are graduated in cubic centimeters.

the pipette. The reagent acts more rapidly on the gas if the water bottle is raised and lowered a few times. This movement of the water in the measuring tube agitates the gas and also the reagent and exposes more of the gas to the direct action of the absorbent. To increase the surface over which the reagents can act, the pipettes are filled with small glass tubes. When the gas has been in the first pipette for about a minute, it should be drawn back into the measuring tube with the level of the reagent brought back to the mark where it was originally, and the cock closed. The pressure of the gas is again made atmospheric and its volume measured. Now repeat this operation until **two or three measurements are obtained which are alike,** showing that all the CO_2 has been absorbed. Then the cock on the tube leading to the pipette containing the absorbent for oxygen can be opened, the gas forced over, and measured several times till a constant volume is observed. Finally the gas is passed into the third pipette for absorbing CO, repeating the operation of measuring as with the other pipettes.

The absorption of oxygen will usually require considerably more time than for the determinations of the carbonic acid (CO_2) and the carbonic oxide (CO), so that it is unnecessary to make a measurement of the volume of the gas till after the gas has been exposed about three minutes to the reagent. Soft rubber bags (see **Fig. 174**) should be attached by means of glass tubes to the corks shown in the pipettes on the farther side in **Fig. 173** and are provided to protect the reagents from absorbing oxygen and carbonic oxide from the air. Both of these reagents will absorb oxygen from atmospheric air, so that the access of fresh air must be prevented. The rubber bags are useful also for the purpose of producing alternately, with the pressure of the hand, suction and pressure for agitating the reagents.

A form of Orsat apparatus particularly suitable for portable use, and in which renewals of broken parts can be cheaply and easily made, is illustrated in **Fig. 175**. This apparatus, designed by Professor John R. Allen and the author,[1] is also particularly suitable for the use of engineers because the pipettes containing the reagents can be removed from the apparatus very easily

[1] Made commercially by the Bausch & Lomb Co., Rochester, N. Y.

for changing solutions. They can be emptied, refilled, and replaced in a very short time. The absorption pipettes (**Fig. 176**) are made simply of two glass test-tubes, the smaller one inside the larger one. The small test-tube is held as it were inverted, and has a glass " capillary " tube fused into its closed end. The outer tube is closed at the top by a rubber stopper through which the capillary portion of the inner tube passes. Very small glass tubes are placed in the inner test-tube to

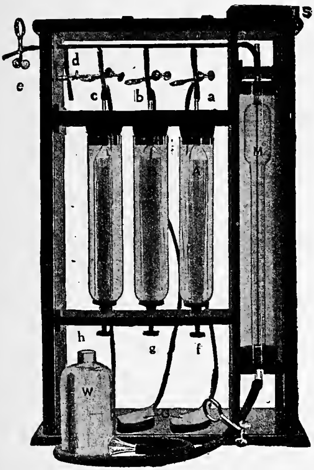


Fig. 175.—Allen-Moyer Gas Apparatus.

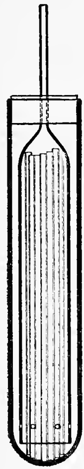


Fig. 176.—Absorption Pipette of Allen-Moyer Gas Apparatus.

increase the surface for the action of the reagent. The pipette is held in place by means of a hard-rubber disk supported on brass screws. The level of the reagent in the pipette is established when the air in the inner tube is drawn out and the level of the liquid rises to a mark on the glass capillary tubes. In the usual forms of the Orsat apparatus the pipettes invariably become leaky at the stopper provided for emptying. In the Allen-Moyer apparatus there is no opportunity for such leakage.

When the sample of the gas is passed through the capillary tube into the inner test-tube, the reagent is displaced

and raises the level in the outer test-tube. Similarly when the gas is passed back into the measuring tube the level falls in the outer tube, rises in the inner one, and is brought back to the original level at the mark on the capillary tube. Otherwise the method of operation is the same as described for Fisher's apparatus (**Fig. 174**).

In this apparatus the measuring tube **M** and water bottle **W** are of the same type as those used in Fisher's design. The yoke is also similar, although usually made of hard rubber to avoid breaking it in transportation. It has also spring pinch-cocks instead of ground-glass cocks. When glass cocks are used by inexperienced persons all sorts of difficulties are likely to result, particularly that it often happens that they are not pressed into their seats tightly enough to prevent the loss of gas or the entrance of air. Sometimes the glass cocks will be put into their seats so tightly that it is impossible to move them without breaking. These difficulties, although met often enough in laboratory work, are still more frequently observed in practice.

Coefficient of Dilution. The coefficient of dilution is the ratio of the volume of the air supplied to the volume theoretically necessary to provide the oxygen required for combustion. It will now be shown how this coefficient can be calculated from an analysis of the flue gases.

Oxygen when combining with carbon to form carbonic dioxide produces a volume equal to itself, thus,

$$\mathbf{C+O_2=CO_2},$$

and in forming carbonic oxide produces twice the volume

$$\mathbf{2C+O_2=2CO}.$$

Now if we use symbols to designate the percentages by volume of the gases in a sample of flue gas as follows:

a is the percentage by volume CO_2,
b is the percentage by volume O_2,
c is the percentage by volume CO,
d is the percentage by volume N (nitrogen).

Then the volume occupied by the free oxygen in the air before combining with the carbon was $\mathbf{a+b+\frac{1}{2}c}$ per cent, while that required is obviously $\mathbf{a+\frac{1}{2}c}$ per cent.

The coefficient of dilution is therefore,

$$\frac{\mathbf{a}+\mathbf{b}+\frac{1}{2}\mathbf{c}}{\mathbf{a}+\frac{1}{2}\mathbf{c}}. \quad \text{. . . . .} \quad (53)$$

In a little different form the reactions given in the last paragraph may be stated (1) for carbon burned to CO_2,

$$\underset{\substack{(2 \text{ vols.})\\ 24}}{2C} + \underset{\substack{(2 \text{ vols.})\\ 64}}{2O_2} = \underset{\substack{(2 \text{ vols.})\\ 88}}{2CO_2}, \quad \text{. . . . . .} \quad (54)$$

and (2) for carbon burned to CO,

$$\underset{\substack{(2 \text{ vols.})\\ 24}}{2C} + \underset{\substack{(1 \text{ vol.})\\ 32}}{O_2} = \underset{\substack{(2 \text{ vols.})\\ 56}}{2CO}. \quad \text{. . . . . .} \quad (55)$$

The ratio of the volume of the carbon vapor burned to CO_2, to the volume burned to CO is the same as the ratio of the volume of CO_2 in the products of combustion is to the volume of CO. Further since the ratio of volumes of the carbon vapor is obviously the same as the ratio of the corresponding weights, we may say that in a mixture of gases, the ratio of the **weight** of carbon required to produce the CO_2 to the **weight** needed for the CO is equal to the ratio of the **volume** of CO_2 in the mixture to the **volume** of CO.

The atomic weights of carbon and oxygen show (equation (54)) that a volume of oxygen is $2\frac{2}{3}$ (64/24) times as heavy as an equal volume of carbon vapor. It follows then that for burning one pound of carbon to CO_2, $2\frac{2}{3}$ pounds of oxygen are required. The other reaction (55) showing the combination of carbon and oxygen to form CO shows with the same reasoning that $1\frac{1}{3}$ pounds of oxygen are required to burn one pound of carbon to CO.

In the general case we are considering and using symbols **a** and **c** respectively as before to represent the volumes of CO_2 and of CO in the flue gases, then the weight of the carbon burned to CO_2 is to the weight burned to CO as **a** is to **c** and if $\frac{\mathbf{a}}{\mathbf{a}+\mathbf{c}}$ represents the weight of carbon burned to CO_2 and $\frac{\mathbf{c}}{\mathbf{a}+\mathbf{c}}$ the

weight burned to CO, then the weight of oxygen required per pound of carbon is $2\frac{2}{3}\left(\frac{a}{a+c}\right)+1\frac{1}{3}\left(\frac{c}{a+c}\right)$ and the weight of air[1] per pound of carbon is in pounds,

$$\frac{100}{23}\left[2\frac{2}{3}\left(\frac{a}{a+c}\right)+1\frac{1}{3}\left(\frac{c}{a+c}\right)\right]. \quad . \quad . \quad . \quad . \qquad (56)$$

If **z** is the percentage by weight of carbon then the weight of air in pounds per pound of coal is

$$\frac{100z}{23}\left[2\frac{2}{3}\left(\frac{a}{a+c}+1\frac{1}{3}\frac{c}{a+c}\right)\right]. \quad . \quad . \quad . \quad . \quad . \qquad (57)$$

Volumetric analyses of the flue gases can be used also to calculate the weight of the products of combustion (flue gases) per pound of coal burned and also the heat units lost in these gases. For this calculation the relations of the molecular weights are important.

Molecular weight of $CO_2=44$
$O_2=32$
$CO=28$
$N_2=28$
$C=12$

Now in a sample of **x** pounds of flue gases in which the percentages by volume are represented by the symbols **a, b, c, d,** the relative percentages by weights of the constituents will be

$$CO_2=\frac{44a}{x};$$

$$O_2=\frac{32b}{x};$$

$$CO=\frac{28c}{x};$$

$$N_2=\frac{28d}{x};$$

and we can write further

[1] Weight of air may be checked with Peabody's and Jacobus' equations (61, 62 and 63[1]), pages 225–228.

Weight of carbon burned to CO_2 in x pounds of gas

$$=\frac{12}{44}\times 44a = 12a.$$

Weight of carbon burned to CO in x pounds of gas

$$=\frac{12}{28}\times 28c = 12c.$$

Total weight of carbon burned in x pounds of gas $= 12(a+c)$.

Total weight of carbon burned per pound of gas $= 12\frac{(a+c)}{x}$.

Total weight of gas generated per pound of carbon $=\frac{x}{12(a+c)}$.

Of this total weight of gas as expressed by the last equation the constituents are distributed in percentages by weight as follows:

Weight of CO_2 in samples per pound carbon burned,

$$w_1 = \frac{44a.x}{12x(a+c)},$$

or we may write

$$CO_2 = w_1 = \frac{44a}{12(a+c)}\text{lb.};$$

and similarly,

$$O_2 = w_2 = \frac{32b}{12(a+c)}\text{lb.};$$

$$CO = w_3 = \frac{28c}{12(a+c)}\text{lb.};$$

$$N_2 = w_4 = \frac{28d}{12(a+c)}\text{lb.}$$

Total weight of gases w_g per pound of coal burned, if there is z **per cent**[1] of carbon in the coal, is in pounds

$$w_g = \frac{z(44a + 32b + 28c + 28d)}{12(a+c)100} \quad . \quad . \quad . \quad . \quad . \quad (58)$$

[1] It may be assumed for very approximate values that $z = 1 - y$ where y is the per cent of ash in the coal.

Now if we represent by t_f and t_a the temperatures respectively in degrees Fahrenheit of the gases in the flue and of the air entering the furnace, then the **heat lost** in the flue gases Q_g per pound of coal is, inserting values of specific heats,[1]

$$Q_g=\frac{z}{100}(.217w_1+.217w_2+.245w_3+.244w_4)(t_f-t_a).$$

The total heat generated Q_0 by the more or less incomplete combustion of one pound of coal when there are a and c percentages by volume respectively of CO_2 and CO in the flue gas is

$$Q_0=\frac{z}{100}\left(\frac{a}{a+c}\times 14,600+\frac{c}{a+c}\times 4400\right)(\text{B.T.U.}). \quad . \quad (59)$$

Since the heat of combustion of carbon when burned to CO_2 is approximately 14,600 and when burned to CO is about 4400 B.T.U.

Finally, if Q_p is the heat from perfect combustion or the "calorific" value in B.T.U. of a pound of coal, then the **efficiency of the furnace**[2] $=\frac{Q_0}{Q_p}$. The percentage of heat from perfect combustion lost in the flue gases $=\frac{Q_g}{Q_p}$.

Recording Apparatus for Determining CO_2. A typical apparatus for making a continuous record of the percentage by volume of carbonic dioxide in gases is shown in **Fig. 177.** The gas is taken to the instrument from the side flue or last combustion chamber of each boiler or furnace to the inlet pipe D and is drawn through the machine by a special water aspirator Q, fixed to the top of the instrument by means of the standard T. After actuating the aspirator Q, a portion of the water flows to the small tank L, which serves as a pressure regulator, and is provided with an overflow tube R. From this tank the water

[1] This method of finding the heat escaping in the flue gases can be used to correct determinations made with the Junkers calorimeter (page 176) when the products of combustion are discharged at a temperature different from that of the room.

[2] For the calculation of related quantities see "Heat Balance" (A.S. M.E. Rules), page 214; also Allen and Bursley's *Heat Engines*, pages 64–65.

enters the tube *H* in a fine stream, which is adjusted by the cock *S* and gradually fills the vessel *K*. This vessel consists of an upper and a lower compartment, the two being in communication through a tube erected in the upper chamber and reaching nearly to the top. Water, which enters this vessel *K* through the tube *H*, gradually fills the upper chamber and thus compresses the air contained in it. This pressure is transmitted to the lower compartment through the communication tube mentioned above, and acts upon the mixture of glycerine and water with which this is filled, driving it out into the calibrated tube *C*. When the rising liquid in *C* has reached the inlet and outlet to this vessel, no more gas can enter the calibrated tubes and the aspirator will now draw the gas through the seal *F*.

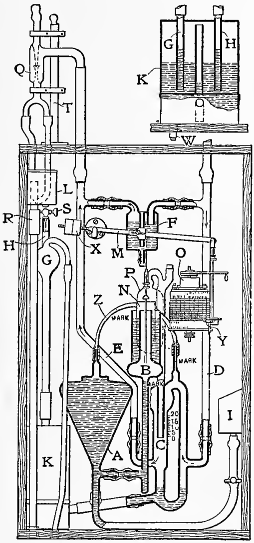


FIG. 177.—Recording CO_2 Apparatus.

Before the liquid can close the central tube in *C*, the gas must overcome the slight resistance offered by the elastic bag *P*, and is thereby forced to assume atmospheric pressure. When the liquid has sealed the lower open end of this central tube, exactly 100 cubic centimeters of flue gas are trapped off in the outer vessel *C* and its companion tube, under atmospheric pressure. As the liquid rises, the gas is forced through the thin tube *Z* into the vessel *A*, which is filled with a solution of caustic potash for absorbing carbonic dioxide.

The gas remaining gradually displaces the potash solution in *A*, sending it up into the vessel *B*. This has an outer jacket, filled with glycerine and supporting a float *N*. Through the center of this float reaches a thin tube, through which the air in *B* is kept at atmospheric pressure. The float is suspended from the pen gear *M* by a silk cord and counter-balanced by the

weight X. The liquid in B forces a portion of the air through the central tube in the float, and then raises the latter, causing the pen lever to swing upward, carrying the pen, Y, with it.

The mechanism is so calibrated and adjusted that the pen will travel to the top, or zero line, on the chart when only atmospheric air is passing through the machine, and nothing is absorbed by the potash in A. When there is any carbonic dioxide in the gas it is absorbed by the potash in A, and not so much of this liquid would be forced up into the vessel B.

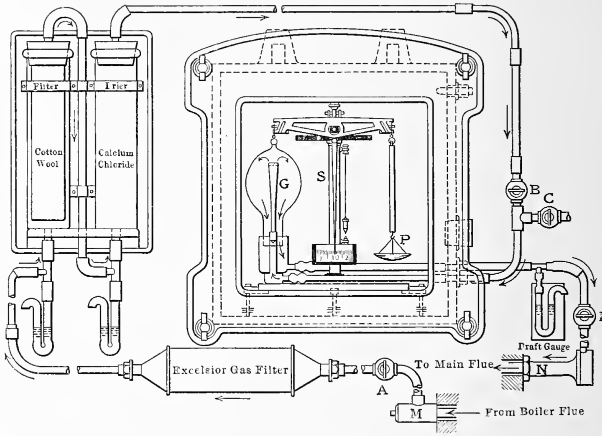


FIG. 178.—CO_2 "Weighing" Apparatus (Econometer).

The float would not then cause the pen to travel up so high on the chart, in proportion to the amount of CO_2 absorbed.

Another apparatus for making continuous determinations of CO_2 in flue gases is shown in **Fig. 178.** Gas from the boiler flue enters at M, passes through an excelsior filter where dust is removed, and then goes on through tubes leading it through glass vessels containing cotton wool and calcium chloride. After being cleaned it passes in the direction of the arrows through the valve B into the weighing apparatus. On account of the greater specific gravity of CO_2 the larger the percentage of this gas the greater the tendency will be to pull downward

the vessel G so that the pointer S on the balance can be adjusted to make the scale over which it travels indicate the percentage of CO_2. For such a method of determination, obviously, the gas must be clean and dry. The cleaning is done by the excelsior and wool filters and the drying is done by the calcium chloride.

CHAPTER X

BOILER TESTING

Tests of steam boilers are made to determine usually the following principal results:

(1) Quantity of steam evaporated or furnished per hour.

(2) Efficiency as a heat user, or weight of water evaporated per pound of combustible (fuel less moisture and ash).

(3) Weight of water evaporated per hour per square foot of water-heating surface.

(4) Weight of fuel burned per hour per square foot of grate surface.

Leakages of any kind are always a lurking enemy for those engaged in any kind of accurate testing, and work with boilers is no exception. Poor results with boilers are due more often to air leakage than to any other fault. Air entering the setting and flues instead of the furnace does not assist combustion, but, on the contrary, absorbs from the hot gases a quantity of heat which otherwise might pass through the boiler-heating surfaces into the water in the boiler. When the object of a series of tests is, for example, to compare one kind of coal with another, or one type of grate or mechanical stoker with another, the losses due to air leakage would not be of much consequence in what are only comparative results; but if a boiler is to give the best possible efficiency and capacity, air leaks must be stopped.

In practice the importance of closing air leaks in the boiler setting is forcefully presented when patented devices for fuel saving are installed in boiler plants. Important economies in many cases are guaranteed if the new device is adopted, and then the claims of the agent are made good by instructing his workmen to go over the boiler, closing up all cracks in the setting through which cold air could enter, and at the same

time covering the outside surface of the setting with a coating impervious to air. By such means the owner of the plant pays a high price for results that could have been obtained much more cheaply.

The first of the principal objects of a boiler test stated above is to determine its capacity "rating."[1] The unit of capacity most generally used in steam boiler practice is the "boiler" horse power. Now this term horse power has two very distinct meanings in engineering practice. Usually it is taken to mean the rate of doing work or the work done in a definite period of time. In this sense it means, as in the case of engines, turbines, water-wheels, etc., 33,000 foot-pounds per minute.[2] In the case of a steam boiler, however, where the work done must be measured by the conversion of water into steam, a horse power is taken as the evaporation of 30 pounds of water at a temperature of 100 degrees Fahrenheit into steam at 70 pounds pressure above the atmosphere. When this unit was adopted it was considered that 30 pounds per hour was approximately the requirement per indicated horse power of an average engine. The Committee on Boiler Tests of the American Society of Mechanical Engineers (page 218) have adopted what is in effect the same unit, stating it, however, somewhat differently—that a **boiler horse power is equivalent to evaporating 34.5 pounds of steam per hour from feedwater temperature of 212 degrees Fahrenheit into steam at the same temperature.** According to the latest steam tables this is equivalent to approximately 33,480 B.T.U. per hour, or 558 B.T.U. per minute.

Unit of Evaporation. For reducing the results of the boiler tests to a common standard the term "unit of evaporation" is used. It is the heat required to evaporate a pound of water

[1] When the steam supply is small or if, for any reason, the noise due to escaping steam from the calorimeters used is objectionable, a calorimeter may be shut off sometimes between readings. The length of time the calorimeter is in operation must then be carefully noted in order to determine the weight of steam lost through it by calculating the flow through its orifice (see page 148). The flow of steam, however, through the calorimeter must always be started before observations of the temperatures are to be taken in order to get **constant conditions.**

[2] This unit of horse power was adopted by James Watt, who considered it equivalent to the work done by a good London draft horse.

from and at 212 degrees Fahrenheit, which according to standard steam tables is approximately equivalent to 970.4 B.T.U.[1]

Graphical Log Sheets of boiler tests similar to the one shown in **Fig. 179** are very serviceable for checking the observations when made during the test as the data are taken. In

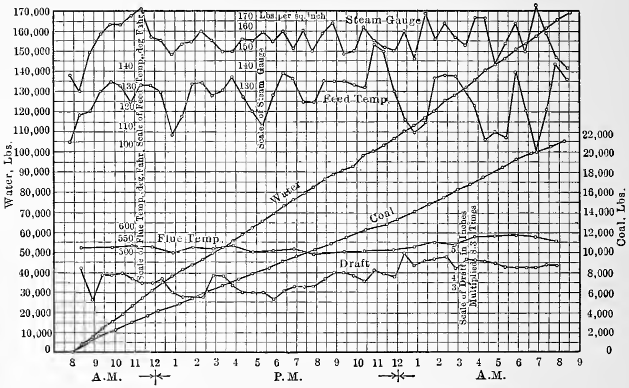


FIG. 179.—Graphical Chart of a Boiler Trial.

the report of a test it shows also the relative irregularity or regularity of the conditions affecting the results.

Standard Methods for Boiler Trials. The American Society of Mechanical Engineers has adopted rules for conducting boiler trials which are generally accepted in America and are also considered with favor in England.[2] These rules are so complete that they will be given here with practically no abridgement.[3]

[1] Marks and Davis' *Steam Tables and Diagrams*, see also *Peabody's Steam Tables*, 1909 Edition.

[2] *Engines and Boilers* by W. W. F. Pullen, pages 466–475.

[3] *Transactions American Society of Mechanical Engineers*, vol. 31, pages 34–111 (including discussions).

RULES FOR CONDUCTING BOILER TRIALS

ABRIDGED CODE OF 1899.

I. *Determine at the outset* the specific object of the proposed trial, whether it be to ascertain the capacity of the boiler, its efficiency as a steam generator, its efficiency and its defects under usual working conditions, the economy of some particular kind of fuel, or the effect of changes of design, proportion or operation; and prepare for the trial accordingly.

II. *Examine the boiler*, both outside and inside; ascertain the dimensions of grates, heating sufaces, and all important parts; and make a full record, describing the same, and illustrating special features by sketches. The area of heating surface is to be computed from the surfaces of shells, tubes, furnaces, and fire boxes in contact with the fire or hot gases. The outside diameter of water tubes and the inside diameter of fire tubes are to be used in the computation. All surfaces below the mean water level which have water on one side and products of combustion on the other are to be considered as water-heating surface, and all surfaces above the mean water level which have steam on one side and products of combustion on the other are to be considered as superheating surface.

III. *Notice the general condition* of the boiler and its equipment, and record such facts in relation thereto as bear upon the objects in view.

If the object of the trial is to ascertain the maximum economy or capacity of the boiler as a steam generator, the boiler and all its appurtenances should be put in first-class condition. Clean the heating surface inside and outside, remove clinkers from the grates and from the sides of the furnace. Remove all dust, soot, and ashes from the chambers, smoke connections, and flues. Close air leaks in the masonry and poorly fitted cleaning doors. See that the damper can be opened wide and closed tightly. Test for air leaks by firing a few shovelsful of smoky fuel and immediately closing the damper, observing the escape of smoke through the crevices, or by passing the flame of a candle over cracks in the brickwork.

IV. *Determine the character of the coal* to be used. For tests of the efficiency or capacity of the boiler for comparison with other boilers the coal should, if possible, be of some kind which is commercially regarded as a standard. For New England and that portion of the country east of the Allegheny Mountains, good anthracite egg coal, containing not over 10 per cent of ash, and semi-bituminous Clearfield (Pa.), Cumberland (Md.), and Pocahontas (Va.) coals are thus regarded. West of the Allegheny Mountains, Pocahontas (Va.) and New River (W. Va.) semi-bituminous, and Youghiogheny or Pittsburg bituminous coals are recognized as standards.[1] There is no special grade of coal mined in the Western States which is widely recognized as of superior quality or considered as a standard coal for boiler testing. Big Muddy lump, an Illinois coal mined in Jackson County, Ill., is suggested as being of sufficiently high grade to answer these requirements in districts where it is more conveniently obtainable than the other coals mentioned above.

For tests made to determine the performance of a boiler with a particular kind of coal, such as may be specified in a contract for the sale of a boiler, the coal used should not be higher in ash and in moisture than that specified, since increase in ash and moisture above a stated amount is apt to cause a falling off of both capacity and economy in a greater measure than the proportion of such increase.

V. *Establish the correctness of all apparatus* used in the test for weighing and measuring. These are:

1. Scales for weighing coal, ashes, and water.

2. Tanks, or water meters for measuring water. Water meters, as a rule, should only be used as a check on other measurements. For accurate work, the water should be weighed or measured in a tank.

3. Thermometers and pyrometers for taking temperatures of air, steam, feed-water, waste gases, etc.

4. Pressure gages, draught gages, etc.

The kind and location of various pieces of testing appa-

[1] These coals are selected because they are about the only coals which possess the essentials of excellence of quality, adaptability of various kinds of furnaces, grates, boilers, and methods of firing, and wide distribution and general accessibility in the markets.

ratus must be left to the judgment of the person conducting the test; always keeping in mind the main object, i.e., to obtain authentic data.

VI. *See that the boiler is thoroughly heated* before the trial to its usual working temperature. If the boiler is new and of a form provided with a brick setting, it should be in regular use at least a week before the trial, so as to dry and heat the walls. If it has been laid off and become cold, it should be worked before the trial until the walls are well heated.

VII. *The boiler and connections* should be proved to be free from leaks before beginning a test, and all water connections, including blow and extra feed pipes, should be disconnected, stopped with blank flanges, or bled through special openings beyond the valves, except the particular pipe through which water is to be fed to the boiler during the trial. During the test the blow-off and feed-pipes should remain exposed to view.

If an injector is used, it should preferably receive steam directly through a felted pipe from the boiler being tested.[1]

If the water is metered after it passes the injector, its temperature should be taken at the point where it leaves the injector. If the quantity is determined before it goes to the injector, the temperature should be determined on the suction side of the injector, and if no change of temperature occurs other than that due to the injector, the temperature thus determined is properly that of the feed water. When the temperature changes between the injector and the boiler, as by the use of a heater or by radiation, the temperature at which the water enters and leaves the injector and that at which it enters the boiler should all be taken. In that case, the weight to be used is that of the water leaving the injector computed from the heat units if not directly measured, and the temperature, that of the water entering the boiler.

[1] In feeding a boiler undergoing test with an injector taking steam from another boiler, or from the main steam pipe from several boilers, the evaporative results may be modified by a difference in the quality of the steam from such source compared with that supplied by the boiler being tested, and in some cases the connection to the injector may act as a drip for the main steam pipe. If it is known that the steam from the main pipe is of the same pressure and quality as that furnished by the boiler undergoing the test, the steam may be taken from such main pipe.

Let $\mathbf{w}$ = weight of water entering the injector, pounds.
$\mathbf{x}$ = weight of steam entering the injector, pounds.
$\mathbf{h}_1$ = heat units per pound of water entering injector.
$\mathbf{h}_2$ = heat units per pound of steam entering injector.
$\mathbf{h}_3$ = heat units per pound of water leaving injector.

Then, $\mathbf{w}+\mathbf{x}$ = weight of water leaving injector.

$$\mathbf{x}=\mathbf{w}\frac{\mathbf{h}_3-\mathbf{h}_1}{\mathbf{h}_2-\mathbf{h}_3}. \qquad (60)$$

See that the steam main is so arranged that water of condensation cannot run back into the boiler.

When coal is used for fuel it is usually weighed in wheelbarrows which can be pushed upon large platform scales. The time when beginning to fire from each loaded wheelbarrow should be noted.

VIII. *Duration of the Test.*—For tests made to ascertain either the maximum economy or the maximum capacity of a boiler, irrespective of the particular class of service for which it is regularly used, the duration should be at least 10 hours of continuous running. If the rate of combustion exceeds 25 pounds of coal per square foot of grate surface per hour, it may be stopped when a total of 250 pounds of coal has been burned per square foot of grate.

In cases where the service requires continuous running for the whole 24 hours of the day, with shifts of firemen a number of times during that period, it is well to continue the test for at least 24 hours.

When it is desired to ascertain the performance under the working conditions of practical running, whether the boiler be regularly in use 24 hours a day or only a certain number of hours out of each 24, the fires being banked the balance of the time, the duration should not be less than 24 hours.

IX. *Starting and Stopping a Test.*—The conditions of the boiler and furnace in all respects should be, as nearly as possible, the same at the end as at the beginning of the test. The steam pressure should be the same; the water level the same; the fire upon the grates should be the same in quantity and condition, and the walls, flues, etc., should be of the same temperature. Two methods of obtaining the desired equality

of conditions of the fire may be used, viz., "the standard method" and "the alternate method," the latter being employed where it is inconvenient to make use of the standard method.[1]

X. *Standard Method of Starting and Stopping a Test.*—Steam being raised to the working pressure, remove rapidly all the fire from the grate, close the damper, clean the ash pit, and as quickly as possible start a new fire with weighed wood and coal, noting the time and the water level[2] while the water is in a quiescent state, just before lighting the fire.

At the end of the test remove the whole fire, which has been burned low, clean the grates and ash pit, and note the water level when the water is in a quiescent state, and record the time of hauling the fire. The water level should be as nearly as possible the same as at the beginning of the test. If it is not the same, a correction should be made by computation, and **not by operating the pump after the test is completed.**

XI. *Alternate Method of Starting and Stopping a Test.*—The boiler being thoroughly heated by a preliminary run, the fires are to be burned low and well cleaned. Note the amount of coal left on the grate as nearly as it can be estimated; note the pressure of steam and the water level. Note the time, and record it as the starting time. Fresh coal which has been weighed should now be fired. The ash pits should be thoroughly cleaned at once after starting. Before the end of the test the fires should be burned low, just as before the start, and the fires cleaned in such a manner as to leave a bed of coal on the grates of the same depth, and in the same condition, as at the start. When this stage is reached, note the time and record it as the stopping time. The water level and steam

[1] The Committee concludes that it is best to retain the designations "standard" and "alternate," since they have become widely known and established in the minds of engineers and in the reprints of the Code of 1885. Many engineers prefer the "alternate" to the "standard" method on account of its being less liable to error due to cooling of the boiler at the beginning and end of a test.

[2] The gage-glass should not be blown out within an hour before the water level is taken at the beginning and end of a test, otherwise an error in the reading of the water level may be caused by a change in the temperature and density of the water in the pipe leading from the bottom of the glass into the boiler.

pressures should previously be brought as nearly as possible to the same point as at the start.

XII. *Uniformity of Conditions.*—In all trials made to ascertain maximum economy or capacity, the conditions should be maintained uniformly constant. Arrangements should be made to dispose of the steam so that the rate of evaporation may be kept the same from beginning to end. This may be accomplished in a single boiler by carrying the steam through a waste steam pipe, the discharge from which can be regulated, as desired. In a battery of boilers, in which only one is tested, the draft may be regulated on the remaining boilers, leaving the test boiler to work under a constant rate of production.

Uniformity of conditions should prevail as to the pressure of steam, the height of water, the rate of evaporation, the thickness of fire, the times of firing and quantity of coal fired at one time, and as to the intervals between the times of cleaning the fires.

The method of firing to be carried on in such tests should be dictated by the expert or person in responsible charge of the test, and the method adopted should be adhered to by the fireman throughout the test.

XIII. *Keeping the Records.*—**Take note of every event connected with the progress of the trial,** however unimportant it may appear. Record the time of every occurrence and the time of taking every weight and every observation.

The coal should be weighed and delivered to the fireman in equal proportions, each sufficient for not more than one hour's run, and a fresh portion should not be delivered until the previous one has all been fired. The time required to consume each portion should be noted, the time being recorded at the instant of firing the last of each portion. It is desirable that at the same time the amount of water fed into the boiler should be accurately noted and recorded, including the height of the water in the boiler, and the average pressure of steam and temperature of feed during the time. By thus recording the amount of water evaporated by successive portions of coal, the test may be divided into several periods if desired, and the degree of uniformity of combustion, evaporation, and economy analyzed for each period. In addition to these records of the coal and the feed-water, not less frequently

than every half hour, observations should be made of the temperature of the feed-water, of the flue gases, of the external air of the boiler room, of the temperature of the furnace when a furnace pyrometer is used, also of the pressure of steam, and of the readings of the instruments for determining the moisture in the steam. A log should be kept on properly prepared blanks containing columns for record of the various observations.

When the " standard method " of starting and stopping the test is used, the hourly rate of combustion and of evaporation and the horse power should be computed from the records taken during the time when the fires are in active condition. This time is somewhat less than the actual time which elapses between the beginning and end of the run. The loss of time due to kindling the fire at the beginning and burning it out at the end makes this course necessary.

XIV. *Quality of Steam.*—The percentage of moisture in the steam should be determined by the use of either a throttling or a separating steam calorimeter. The sampling nozzle should be placed in the vertical steam pipe rising from the boiler. It should be made of one-half-inch pipe, and should extend across the diameter of the steam pipe to within half an inch of the opposite side, being closed at the end and perforated with not less than twenty one-eighth-inch holes equally distributed along and around its cylindrical surface, but none of these holes should be nearer than one-half inch to the inner side of the steam pipe. The calorimeter and the pipe leading to it should be well covered with felting. Whenever the indications of the throttling or separating calorimeter show that the percentage of moisture is irregular, or occasionally in excess of three per cent, the results should be checked by a steam separator placed in the steam pipe as close to the boiler as convenient, with a calorimeter in the steam pipe just beyond the outlet from the separator. The drip from the separator should be caught and weighed, and the percentage of moisture computed therefrom added to that shown by the calorimeter.

Superheating should be determined by means of a thermometer placed in a mercury well inserted in the steam pipe. The degree of superheating should be taken as the difference between the reading of the thermometer for superheated steam and the readings of the same thermometer for saturated steam

at the same pressure as determined by a special experiment, and not by reference to steam tables.

For calculations relating to quality of steam and corrections for quality of steam, see pages 46–60.

XV. *Sampling the Coal and Determining its Moisture.*—As each barrow load or fresh portion of coal is taken from the coal pile, an average shovelful is selected from it and placed in a barrel or box in a cool place and kept until the end of the trial. The samples are then mixed and broken into pieces not exceeding 1 inch in diameter, and reduced by the process of repeated quartering and crushing until a final sample weighing about 5 pounds is obtained, and the size of the larger pieces is such that they will pass through a sieve with one-quarter-inch meshes. From this sample two one-quart air-tight glass preserving jars, or other air-tight vessels which will prevent the escape of moisture from the sample, are to be promptly filled, and these samples are to be kept for subsequent determinations of moisture and of heating value and for chemical analyses. During the process of quartering, when the sample has been reduced to about 100 pounds, a quarter to a half of it may be taken for an approximate determination of moisture. This may be made by placing it in a shallow iron pan, not over 3 inches deep, carefully weighing it, and setting the pan in the hottest place that can be found on the brickwork of the boiler setting of flues, keeping it there for at least 12 hours, and then weighing it. The determination of moisture thus made is believed to be **approximately accurate for anthracite and semi-bituminous coals,** and also for Pittsburg or Youghiogheny coal; but it **cannot be relied upon for coals mined west of Pittsburg, or for other coals containing inherent moisture.** For these latter coals it is important that a more accurate method be adopted. The method recommended by the Committee **for all accurate tests,** whatever the character of the coal, is described as follows:

Take one of the samples contained in the glass jars, and subject it to a thorough air-drying, by spreading it in a thin layer and exposing it for several hours to the atmosphere of a warm room, weighing it before and after, thereby determining the quantity of surface moisture it contains. Then crush the whole of it by running it through an ordinary coffee mill

adjusted so as to produce somewhat coarse grains (less than one-sixteenth-inch), thoroughly mix the crushed sample, select from it a portion of from 10 to 50 grams, weigh it in a balance which will easily show a variation as small as 1 part in 1000, and dry it in an air or sand bath at a temperature between 240 and 280 degrees Fahrenheit for one hour. Weigh it and record the loss, then heat and weigh it again repeatedly, at intervals of an hour or less, until the minimum weight has been reached and the weight begins to increase by oxidation of a portion of the coal. The difference between the original and the minimum weight is taken as the moisture in the air-dried coal. This moisture test should preferably be made on duplicate samples, and the results should agree within 0.3 to 0.4 of one per cent, the mean of the two determinations being taken as the correct result. The sum of the percentage of moisture thus found and the percentage of surface moisture previously determined is the **total moisture.**

XVI. *Treatment of Ashes and Refuse.*—The ashes and refuse are to be weighed in a dry state. If it is found desirable to show the principal characteristics of the ash, a sample should be subjected to a proximate analysis and the actual amount of incombustible material determined. For elaborate trials a complete analysis of the ash and refuse should be made.

XVII. *Calorific Tests and Analysis of Coal.*—The quality of the fuel should be determined either by heat test or by analysis, or by both.

The rational method of determining the total heat of combustion is to burn the sample of coal in an atmosphere of oxygen gas, the coal to be sampled as directed in Article XV. of this code.

The chemical analysis of coal should be made only by an expert chemist. The total heat of combustion computed from the results of the ultimate analysis may be obtained by the use of Dulong's formula (with constants modified by recent determinations), viz., $14{,}600C+62{,}000\left(H-\frac{O}{8}\right)+4000S$, in which C, H, O, and S refer to the proportions of carbon, hydrogen, oxygen, and sulphur respectively, as determined by the ultimate analysis.[1]

[1] Favre and Silberman give 14,544 B.T.U. per pound carbon; Berthe-

It is desirable that a proximate analysis should be made, thereby determining the relative proportions of volatile matter and fixed carbon. These proportions furnish an indication of the leading characteristics of the fuel, and serve to fix the class to which it belongs. As an additional indication of the characteristics of the fuel, the specific gravity should be determined.

XVIII. *Analyses of Flue Gases.*—The analysis of the flue gases is an especially valuable method of determining the relative value of different methods of firing, or of different kinds of furnaces. In making these analyses great care should be taken to procure average samples—since the composition is apt to vary at different points of the flue.

The composition is also apt to vary from minute to minute, and for this reason the drawings of gas should last a considerable period of time. Where complete determinations are desired, the analyses should be intrusted to an expert chemist. For approximate determinations "Orsat"[1] apparatus may be used by the engineer.

For the continuous indication of the amount of carbonic acid present in the flue gases, an instrument may be employed which shows the weight of the sample of gas passing through it.

XIX. *Smoke Observations.*—It is desirable to have a uniform system of determining and recording the quantity of smoke produced where bituminous coal is used. The system commonly employed is to express the degree of smokiness by means of percentages dependent upon the judgment of the observer. The Committee does not place much value upon a percentage method, because it depends so largely upon the personal element, but if this method is used, it is desirable that, so far as possible, a definition be given in explicit terms, as to the basis and method employed in arriving at the percentage.

XX. *Miscellaneous.*—In tests for purposes of scientific research, in which the determination of all the variables entering into the test is desired, certain observations should

lot 14,647 B.T.U. Favre and Silberman give 62,032 B.T.U. per pound hydrogen; Thomsen 61,816 B.T.U.

[1] See R. S. Hale's paper on "Flue Gas Analysis *Transactions A.S.M.E.*, vol. 18., page 901.

be made which are in general unnecessary for ordinary tests. These are the measurement of the air supply, the determination of its contained moisture, the determination of the amount of heat lost by radiation, of the amount of infiltration of air through the setting, and (by condensation of all the steam made by the boiler) of the total heat imparted to the water.

XXI. *Calculations of Efficiency.*—Two methods of defining and calculating the efficiency of a boiler are recommended. They are:

1. Efficiency of the boiler $=\dfrac{\text{Heat absorbed per lb. combustible}}{\text{Calorific value of 1 lb. combustible}}$;

2. Efficiency of the boiler and grate $=\dfrac{\text{Heat absorbed per lb. coal}}{\text{Calorific value of 1 lb. coal}}$.

The first of these is sometimes called the efficiency based on combustible, and the second efficiency based on coal. The first is recommended as a standard of comparison for all tests, and this is the one which is understood to be referred to when the word "efficiency" alone is used without qualification. The second, however, should be included in a report of a test, together with the first, whenever the object of the test is to determine the efficiency of the boiler and furnace together with the grate (or mechanical stoker), or to compare different furnaces, grates, fuels, or methods of firing.

The heat absorbed per pound of combustible (or per pound coal) is to be calculated by multiplying the equivalent evaporation from and at 212 degrees per pound combustible (or coal) by 965.7.[1]

XXII. *The Heat Balance.*—An approximate "heat balance," or statement of the distribution of the heating value of the coal among the several items of heat utilized and heat lost may be included in the report of a test when analyses of the fuel and of the chimney gases have been made. It should be reported in the following form:

[1] This value is the one given in the accepted steam tables in 1899 when this code was published. According to the more recent determinations it should be 970. (See Marks and Davis' and Peabody's revised Steam Tables.)

Heat Balance, or Distribution of the Heating Value of the Combustible.

Total Heat Value of 1 lb. of Combustible..............B.T.U.

	B.T.U.	Per Cent.
1. Heat absorbed by the boiler = evaporation from and at 212 degrees per pound of combustible $\times 965.7$.[1]		
2. Loss due to moisture in coal = per cent. of moisture referred to combustible $\div 100 \times [(212-t)+966^{1}+0.48(T-212)]$ (t = temperature of air in the boiler-room, T = that of the flue gases).		
3. Loss due to moisture formed by the burning of hydrogen = per cent. of hydrogen to combustible $\div 100 \times 9 \times [212(-t)+966^{1}+0.48(T-212)]$.		
4.[2] Loss due to heat carried away in the dry chimney gases = weight of gas per pound of combustible $\times 0.24 \times (T-t)$.		
5.[3] Loss due to incomplete combustion of carbon = $\frac{CO}{CO_2+CO} \times \frac{\text{per cent C in combustible}}{100} \times 10{,}150$.		
6. Loss due to unconsumed hydrogen and hydrocarbons, to heating the moisture in the air, to radiation, and unaccounted for. (Some of these losses may be separately itemized if data are obtained from which they may be calculated).		
Totals............................		100.00

[1] This value is the one given in the accepted steam tables in 1899 when this code was published. According to the more recent determinations it should be 970. (See Marks and Davis' and Peabody's revised Steam Tables.)

[2] The weight of gas per pound of carbon burned may be calculated from the gas analyses as follows:

Dry gas per pound carbon $= \frac{11CO_2 + 8O + 7(CO+N)}{3(CO_2+CO)}$, in which CO_2, CO, O and N are the percentages by volume of the several gases. As the sampling and analyses of the gases in the present state of the art are liable to considerable errors, the result of this calculation is usually only an approximate one. The heat balance itself is also only approximate for this reason, as well as for the fact that it is not possible to determine accurately the percentage of unburned hydrogen or hydrocarbons in the flue gases.

The weight of dry gas per pound of combustible is found by multiplying the dry gas per pound of carbon by the percentage of carbon in the combustible, and dividing by 100.

[3] CO_2 and CO are respectively the percentage by volume of carbonic acid and carbonic oxide in the flue gases. The quantity 10,150 = Number of heat units generated by burning to carbonic acid one pound of carbon contained in carbonic oxide.

XXIII. *Report of the Trial.*—The data and results should be reported in the manner given in either one of the two following tables, omitting lines where the tests have not been made as elaborately as provided for in such tables. Additional

lines may be added for data relating to the specific object of the test. The extra lines should be classified under the headings provided in the tables, and numbered as per preceding line, with sub letters *a*, *b*, etc. The Short Form of Report, Table No. 2, is recommended for commercial tests and as a convenient form of abridging the longer form for publication when saving of space is desirable. For elaborate trials, it is recommended that the full log of the trial be shown graphically, by means of a chart. **(Fig. 179.)**

TABLE NO. 1.

Data and Results of Evaporative Test.

Arranged in accordance with the Complete Form advised by the Boiler Test Committee of the American Society of Mechanical Engineers. Code of 1899.

Made by................of................boiler at..............to determine...
....
Principal conditions governing the trial...........................
....
....
Kind of fuel[1]..
Kind of furnace..
State of the weather...

Method of starting and stopping the test ("standard" or "alternate," Art. X. and XI., Code)..

1. *Date of trial*...
2. *Duration of trial*.................................. hours.

Dimensions and Proportions.

(A complete description of the boiler, and drawings of the same if of unusual type, should be given on an annexed sheet.)

3. *Grate surface......width......length......area*.... sq. ft.
4. Height of furnace.............................. ins.
5. Approximate width of air spaces in grate .... in.
6. Proportion of air space to whole grate surface....... per cent.
7. *Water-heating surface*............................ sq. ft.
8. *Superheating surface*............................ "
9. Ratio of water-heating surface to grate surface..... — to 1.
10. Ratio of minimum draft area to grate surface....... 1 to —.

Average Pressures.

11. *Steam pressure by gage*..........................lbs. per sq. in.
12. *Force of draft between damper and boiler*. ....ins. of water
13. Force of draft in furnace........................ "
14. Force of draft or blast in ash pit................. "

[1] The items printed in italics correspond to the items in the "Short Form of Code"

Average Temperatures.

15. Of external air	deg.	....
16. Of fireroom	"	....
17. Of steam	"	....
18. Of feed water entering heater	"	....
19. Of feed water entering economizer	"	....
20. *Of feed water entering boiler*	"	....
21. *Of escaping gases from boiler*	"	....
22. Of escaping gases from economizer	"	....

Fuel.

23. Size and condition	
24. Weight of wood used in lighting fire	lbs.
25. *Weight of coal as fired* [1]	"
26. *Percentage of moisture in coal* [2]	per cent.
27. *Total weight of dry coal consumed*	lbs.
28. *Total ash and refuse*	"
29. Quality of ash and refuse	
30. Total combustible consumed	lbs.
31. *Percentage of ash and refuse in dry coal*	per cent.

Proximate Analysis of Coal.

	Of Coal.	Of Combustible
32. Fixed carbon	per cent.	......per cent.
33. Volatile matter	"	...... "
34. Moisture	"	
35. Ash	"	
	100 per cent.	100 per cent.
36. Sulphur, separately determined	"	"

Ultimate Analysis of Dry Coal.

(Art. XVII., Code.)

	Of Coal.	Of Combustible.
37. Carbon (C)	per cent	......per cent.
38. Hydrogen (H)	"	...... "
39. Oxygen (O)	"	...... "
40. Nitrogen (N)	"	...... "
41. Sulphur (S)	"	...... "
42. Ash	"	
	100 per cent	100 per cent.
43. Moisture in sample of coal as received	"	

[1] Including equivalent of wood used in lighting the fire, not including unburnt coal withdrawn from furnace at times of cleaning and at end of test. One pound of wood is taken to be equal to 0.4 pound of coal, or, in case greater accuracy is desired, as having a heat value equivalent to evaporation of 6 pounds of water from and at 212 degrees per pound. ($6 \times 965.7 = 5794$ B.T.U.) The term "as fired" means in its actual condition, including moisture.

[2] This is the total moisture in the coal as found by drying it artificially, as described in Art. XV. of Code.

Analysis of Ash and Refuse.

44. Carbon	per cent.
45. Earthy matter	"

Fuel per Hour.

46. *Dry coal consumed per hour*	lbs.
47. Combustible consumed per hour	"
48. *Dry coal per square foot of grate surface per hour*	"
49. Combustible per square foot of water-heating surface per hour	"

Calorific Value of Fuel.

(Art. XVII., Code.)

50. *Calorific value by oxygen calorimeter, per lb. of dry coal*	B.T.U.
51. *Calorific value by oxygen calorimeter, per lb. of combustible*	"
52. Calorific value by analysis, per lb. of dry coal [1]	"
53. Calorific value by analysis, per lb. of combustible	"

Quality of Steam.

54. *Percentage of moisture in steam*	per cent.
55. *Number of degrees of superheating*	deg.
56. Quality of steam (dry steam = unity).	

Water.

57. *Total weight of water fed to boiler* [2]	lbs.
58. Equivalent water fed to boiler from and at 212 degrees	"
59. *Water actually evaporated, corrected for quality of steam*	"
60. Factor of evaporation [3]	"
61. Equivalent water evaporated into dry steam from and at 212 degrees [4] (Item 59 × Item 60)	"

Water per Hour.

62. *Water evaporated per hour, corrected for quality of steam*	"
63. *Equivalent evaporation per hour from and at 212 degrees*[2]	"
64. *Equivalent evaporation per hour from and at 212 degrees per square foot of water-heating surface*[2]	"

[1] See formula for calorific value under Article XVII. of Code.

[2] Corrected for inequality of water level and of steam pressure at beginning and end of test.

[3] Factor of evaporation $= \frac{H-h}{965.7}$, in which H and h are respectively the total heat in steam of the average observed pressure, and in water of the average observed temperature of the feed. According to the newer steam tables the divisor in this fraction should be 970.

[4] The symbol "U. E." meaning "Units of Evaporation," may be conveniently substituted for the expression "Equivalent water evaporated into dry steam from and at 212 degrees," its definition being given in a footnote.

Horse Power.

65. *Horse power developed.* ($34\frac{1}{2}$ *lbs. of water evaporated per hour into dry steam from and at 212 degrees, equals one horse power*)[1]	H. P.
66. *Builder's rated horse power*........................	"
67. *Percentage of builders' rated horse power developed*....	per cent.

Economic Results.

68. *Water apparently evaporated under actual conditions per pound of coal as fired.* (*Item* 57 ÷ *Item* 25)....	lbs.
69. *Equivalent evaporation from and at 212 degrees per pound of coal as fired.*[2] (*Item* 61 ÷ *Item* 25.)	"
70. *Equivalent evaporation from and at 212 degrees per pound of dry coal.*[2] (*Item* 61 ÷ *Item* 27.)..........	"
71. *Equivalent evaporation from and at 212 degrees per pound of combustible.*[2] (*Item* 61 ÷ *Item* 30.).......	"

(If the equivalent evaporation, Items 69, 70, and 71, is not corrected for the quality of steam, the fact should be stated.)

Efficiency.

(Art. XXI., Code.)

72. *Efficiency of the boiler; heat absorbed by the boiler per lb. of combustible divided by the heat value of one lb. of combustible* [2]	per cent.
73. *Efficiency of boiler, including the grate; heat absorbed by the boiler, per lb. of dry coal, divided by the heat value or one lb. of dry coal*......................	"

Cost of Evaporation.

74. *Cost of coal per ton of —— lbs. delivered in boiler room*	$
75. Cost of fuel for evaporating 1,000 lbs. of water under observed conditions	$
76. *Cost of fuel used for evaporating 1,000 lbs. of water from and at 212 degrees*............................	$

Smoke Observations.

77. Percentage of smoke as observed..................	per cent.
78. Weight of soot per hour obtained from smoke meter.	ounce.
79. Volume of soot per hour obtained from smoke meter	cu. in.

[1] Held to be the equivalent of 30 lbs. of water per hour evaporated from 100 degrees Fahrenheit into dry steam at 70 lbs. gage pressure. (See Introduction to Code, page 201.)

[2] In all cases where the word combustible is used, it means the coal without moisture and ash, but including all other constituents. It is the same as what is called in Europe "coal dry and free from ash."

Methods of Firing.

80. Kind of firing (spreading, alternate, or coking)......
81. Average thickness of fire........................
82. Average intervals between firings for each furnace during time when fires are in normal condition.....
83. Average interval between times of leveling or breaking up.................................

Analyses of the Dry Gases.

84. Carbon dioxide (CO_2)............................	per cent.
85. Oxygen (O)....................................	"
86. Carbon monoxide (CO)..........................	"
87. Hydrogen and hydrocarbons......................	"
88. Nitrogen (by difference) (N).....................	"
	100 per cent.

TABLE NO. 2.

Data and Results of Evaporative Test.

Arranged in accordance with the Short Form advised by the Boiler Test Committee of the American Society of Mechanical Engineers. Code of 1899.

Made by............on...........boiler, at....................to determine...
Kind of fuel...
Kind of furnace..
Method of starting and stopping the test (" standard " or " alternate," Art. X. and XI., Code)............................

Grate surface..................................	sq. ft.
Waterheating surface...........................	"
Superheating surface............................	"

Total Quantities.

1. Date of trial..................................	
2. Duration of trial..............................	hours.
3. Weight of coal as fired [1].......................	lbs.
4. Percentage of moisture in coal [1]..................	per cent.
5. Total weight of dry coal consumed...............	lbs.
6. Total ash and refuse...........................	"
7. Percentage of ash and refuse in dry coal...........	per cent.
8. Total weight of water fed to the boiler [1]...........	lbs.
9. Water actually evaporated, corrected for moisture or superheat in steam..........................	"
10. Equivalent water evaporated into dry steam from and at 212 degrees [1]..............................	"

[1] See footnotes of Complete Form, pages 216 and 217.

Hourly Quantities.

11.	Dry coal consumed per hour......................	lbs.
12.	Dry coal per square foot of grate surface per hour...	"
13.	Water evaporated per hour corrected for quality of steam.....................................	"
14.	Equivalent evaporation per hour from and at 212 degrees [1]....................................	"
15.	Equivalent evaporation per hour from and at 212 degrees per square foot of water-heating surface [1].	"

Average Pressures, Temperatures, etc.

16.	Steam pressure by gage	lbs. per sq. in.
17.	Temperature of feed water entering boiler..........	deg.
18.	Temperature of escaping gases from boiler..........	"
19.	Force of draft between damper and boiler..........	ins. of water.
20.	Percentage of moisture in steam, or number of degrees of superheating..............................	per cent. or deg.

Horse Power.

21.	Horse power developed (Item $14 \div 34\frac{1}{2}$) [1]...........	H. P.
22.	Builder's rated horse power......................	"
23.	Percentage of builder's rated horse power developed.	per cent.

Economic Results.

24.	Water apparently evaporated under actual conditions per pound of coal as fired. (Item $8 \div$ Item 3).....	lbs.
25.	Equivalent evaporation from and at 212 degrees per pound of coal as fired.[1] (Item $10 \div$ Item 3)......	"
26.	Equivalent evaporation from and at 212 degrees per pound of dry coal.[1] (Item $10 \div$ Item 5)..........	"
27.	Equivalent evaporation from and at 212 degrees per pound of combustible.[1] [Item $10 \div$ (Item $5 -$ Item 6)]..	"
	(If Items 25, 26, and 27 are not corrected for quality of steam, the fact should be stated.)	

Efficiency.

28.	Calorific value of the dry coal per pound............	B.T.U.
29.	Calorific value of the combustible per pound........	"
30.	Efficiency of boiler (based on combustible) [1]........	per cent.
31.	Efficiency of boiler, including grate (based on dry coal)	"

Cost of Evaporation.

32.	Cost of coal per ton of —— lbs delivered in boiler room	$
33.	Cost of coal required for evaporating 1000 pounds of water from and at 212 degrees..................	$

[1] See footnotes of Complete Form, pages 217 and 218.

EFFICIENCY OF THE BOILER

The efficiency of the boiler, including the grate, or the efficiency based on coal, is the quotient arising from dividing heat absorbed by the boiler by the heating value of the total amount of coal supplied to the boiler, including the coal which falls through the grate. It may be conveniently calculated by multiplying the number of pounds of water evaporated from and at 212 degrees Fahrenheit into dry steam per pound of dry coal by 965.7,[1] and dividing the product by the heating value in B. T. U. of one pound of dry coal.

The efficiency of the boiler, not including the grate, or the efficiency based on combustible, is the quotient arising from dividing the heat absorbed by the boiler by the heating value of the combustible burned. It may be calculated by multiplying the number of pounds of water evaporated from and at 212 degrees Fahrenheit into dry steam per pound of combustible by 965.7,[1] and dividing the product by the heating value in B. T. U. of one pound of combustible; the term "combustible" being defined as coal dry and free from ash, or the coal supplied to the boiler less its moisture and the ash and unburned coal which falls through the grate or is otherwise withdrawn from the furnace.

The efficiency of the boiler, not including the grate (or the efficiency based upon combustible) is a **more accurate measure of comparison of different boilers** than the efficiency including the grate (or the efficiency based upon coal), for the latter is subject to a number of variable conditions, such as size and character of the coal, air-spaces between the grate bars, skill of the fireman in saving coal from falling through the grate, etc. It is, moreover, subject to errors of sampling the coal for drying and for analysis, which affect the result to a greater degree than they do the efficiency based upon combustible, for the reason that the heating value per pound of combustible of any sample selected from a given lot (such as a carload) of coal is practically a constant quantity and is independent of the percentage of moisture and ash in the sample; while the sample itself, upon the heating value of which the efficiency based on coal is calculated, may differ in its per-

[1] See page 201 and footnote, page 213.

centage of moisture and ash from the average coal used in the boiler test.

When the object of a boiler test is to determine its efficiency as an absorber of heat, or to compare it with other boilers, the efficiency based on combustible is the one which should be used; but when the object of the test is to determine the efficiency of the combination of the boiler, the furnace, and the grate, the efficiency based on coal must necessarily be used.

It has been proposed that in reporting the efficiency of a boiler when the fuel used contains hydrogen, the efficiency should be considered to be the sum of the percentage of the heating value of the fuel which is utilized by the boiler in making steam of the percentage of that heating value which is lost in the shape of latent heat in the moisture in the chimney gases, which moisture is formed by the burning of the hydrogen. This latent heat may amount to over three per cent of the total heating value of the fuel. The reason assigned for this proposal is that, since it is impossible for this heat to be utilized by the boiler because the gases are discharged at a temperature above 212 degrees Fahrenheit, it should not be charged against the boiler. It is not considered advisable that this method of reporting the efficiency should be adopted (1) because it is opposed to the generally accepted definition of efficiency, which is the useful work received from an apparatus divided by the work (or heat value of the fuel) put into it; (2) because in order to calculate it, it is necessary to know both the percentage of hydrogen in the coal and whether or not all of this hydrogen has been burned to H_2O, the first requiring an analysis of the coal, which is not always obtainable, and the second an analysis of the gases for hydrogen, which cannot be obtained with any approximation to accuracy with our present methods of sampling and analyzing gases; and (3) because it is opposed to the almost universal custom of reporting boiler tests. It is true that the latent heat of the water in the chimney gases cannot be utilized (unless an economizer which discharges its gases below 212 degrees is used), and it is not the fault of the boiler that it cannot be utilized. It may be considered the misfortune of the boiler, when tested with hydrogenous coal, similar to the misfortune under which

an engine labors when it is tested while supplied with a condenser which gives a vacuum of less than 30 inches of mercury. The engine might give a higher efficiency with a vacuum of 30 inches than it would with one of 27 or 28 inches; but it is not customary to credit the engine with the efficiency which it loses on account of the imperfect vacuum.

Since it is well understood that a boiler cannot show quite as high an efficiency (as commonly defined) when using bituminous coal high in hydrogen as when using anthracite nearly free from hydrogen, no harm is done, and much confusion is avoided, by reporting the efficiency as the percentage of the heating value of the coal which is actually utilized in making steam. The fact that bituminous coal is used is always stated in the report of a test made with that coal. If desired, a statement may also be made in the "heat balance" of the approximate or estimated percentage of heat which is lost in the latent heat of the moisture in the chimney gases, together with the loss due to moisture in the coal.

DISTRIBUTION OF THE HEATING VALUE OF THE FUEL

In the operation of a steam boiler the following distribution of the total heating value of the fuel takes place:

1. Loss of coal or coke through the grate.

2. Unburned coal or coke carried in the shape of dust or sparks beyond the bridge wall.

3. Heating to 212 degrees the moisture in the coal, evaporating it at that temperature, and evaporating the steam made from it to the temperature of the flue gases = weight of the moisture in pounds $\times [(212 \text{ degrees} - t) + 966$[1] $+ 0.48(T - 212)]$, in which T is the temperature (Fahrenheit) of the flue gases and t the temperature of the external air.

4. Loss of heat due to steam which is formed by burning the hydrogen contained in the coal, and which passes into the chimney as superheated steam = 9 times the weight of the hydrogen $\times [(212 - t) + 966$[1] $+ 0.48(T - 212)]$.

5. Superheating the moisture in the air supplied to the furnace to the temperature of the flue gases = weight of the moisture $\times 0.48(T - t)$.

[1] See footnote, page 213.

6. Heating of the gaseous products of combustion (not including steam) to the temperature of the flue = their weight $\times 0.24(T-t)$.

7. Loss due to imperfect burning of the carbon of the coal and to non-burning of the volatile gases.

8. Radiation from the boiler and furnace.

9. Heat absorbed by the boiler, or useful work.

Item 1 depends upon the size of the spaces between the grate bars; upon the kind of grate, as a plain, shaking, or traveling grate; upon the size of the coal; upon the character of the coal, as it requires to be more or less distributed on the grate in order to get a sufficient supply of air through it; upon the rate of driving of the furnace, rapid driving with some coals requiring more frequent shaking or cleaning of the grate than slow driving; and upon the skill of the fireman.

Item 2 depends upon the nature and fineness of the coal and upon the force of the draft. It is usually so small as to be inappreciable in its effects upon the results of the trial of a stationary boiler driven with natural draft, but in locomotives, with rapid rates of combustion, it often becomes quite important.

Item 3 depends upon the amount of moisture in the coal.

Item 4 depends upon the amount of hydrogen in the coal.

Item 5 depends upon the amount of moisture in the air. The moisture in the air may be obtained from its temperature and relative humidity, as determined by a wet-and-dry bulb thermometer by reference to hygrometric tables. The loss of heat due to the moisture in the air will rarely exceed 0.25 per cent. of the heating value of the fuel, and it may usually, therefore, be neglected. (For hygrometric tables see page 403.)

Item 6 depends chiefly upon the type and proportions of the boiler, and upon the rate at which it is driven. This item is usually the largest of all the heat losses.

Items 3, 4, 5, and 6 depend also on the temperature of the flue gases.

Item 7 depends upon the character of the coal and of the furnace, and upon the skill of the fireman. This loss may be very large, 20 per cent or more of the heating value of the coal, when highly bituminous coals are used in a furnace not adapted to them.

Item 8 depends chiefly upon the type, size, and setting of the boiler, and, when expressed as a percentage of the total heat of the fuel, upon the rate at which it is driven.

Item 9 is the heat absorbed by the boiler, or the useful work. It is also the difference between the total heating value of the coal and of the sum of the losses of items 1 to 8 inclusive.

COMPUTATION OF THE WEIGHT OF THE CHIMNEY GASES FROM THE ANALYSIS BY VOLUME OF THE DRY GAS.

Two methods of calculating from the analysis by volume of the dry chimney gases the number of pounds of dry chimney gases per pound of carbon, or the weight of air supplied per pound of carbon, have been given by different writers. These may be expressed in the shape of formulæ as follows:

$$\text{(A) Pounds dry gas per pound C} = \frac{11CO_2 + 8O + 7(CO + N)}{3(CO_2 + CO)} \qquad (61)$$

$$\text{(B) Pounds air per pound C} = 5.8\frac{2(CO_2 + O) + CO}{CO_2 + CO}. \qquad (62)$$

Formula A may be derived from the method of computation given in Mr. R. S. Hale's paper on "Flue-Gas Analyses," *Transactions American Society of Mechanical Engineers*, vol. 18, page 902, and formula B from the method given in Peabody's and Miller's "Treatise on Steam Boilers." Both are based on the principle that the density, relatively to hydrogen, of an elementary gas (O and N) is proportional to its atomic weight, and that of a compound gas (CO and CO_2) to one-half its molecular weight. Both formulæ are very nearly accurate when pure carbon is the fuel burned, but formula B is inaccurate when the fuel contains hydrogen, for the reason that that portion of the oxygen of the air supply which is required to burn the hydrogen is contained in the chimney gas as H_2O, and does not appear in the analysis of the dry gas.

The following calculations of a supposed case of combustion of hydrogenous fuel illustrates the accuracy of formula A and the inaccuracy of formula B. Assume that the coal has the

following analysis: C, 66.50; H, 4.55; O, 8.40; N, 1.00; water, 10.00; ash and sulphur, 9.55—total, 100. Assume also that one-tenth of the C is burned to CO, and nine-tenths to CO_2; that the air supply is 20 per cent in excess of that required for this combustion; that the air contains 1 per cent by weight of moisture; and that the S in the coal may be considered as part of the ash. We then have the following synthesis of results of the combustion of **100 pounds of coal:**

	O from Air	$N = O \times \frac{77}{23}$	Total Air	CO_2	CO	HO_2
59.85 lbs. C to $CO_2 \times 2\frac{2}{3}$	159.60	534.31	693.91	219.45	......	.....
6.65 " C to $CO \times 1\frac{1}{3}$	8.87	29.70	38.57	......	15.52	.....
3.50 " H to $H_2O \times 8$	28.00	93.74	121.74	......	......	31.50
	196.47	657.75	854.22	......	......	.....
1.05 " H to H_2O 8.40 " H to H_2O	......	......	......	......	......	9.45
10.00 " Water	......	......	......	......	......	10.00
1.00 " N	......	1.00	......	......	......	.....
9.55 " Ash and S	......	......	......	......	......	.....
100.00	......	......	......	......	......	.....
Excess of air 20 per cent.	39.29	131.55	170.84	......	......	.....
	......	......	1025.06	......	......	.....
Moisture in air 1 per cent.	......	......	......	......	......	10.25
Total wt. gases, 1125.67 lbs.= Total dry gases, 1064.56 lbs.	39.29	790.30	......	219.45	15.52	61.20
Per Cent.	O	N		CO_2	CO	
Total dry gases, by weight,	3.69	74.24	.....	20.61	1.546	.....
Total dry gases, by volume,	3.508	80.656	.....	14.252	1.584	.....

Total gases 1125.76 + ash and S 9.55 = 1135.31 lbs. total products.
Total air 1025.06 + moisture in air 10.25 + coal 100 = 1135.31 lbs.
Dry gas per lb. coal 10.6456; per lb. carbon = 10.6456 ÷ .665 = 16.008 lbs.
Dry air per lb. coal 10.2506; per lb. carbon = 10.2506 ÷ .665 = 15.414 lbs.

Computation of the weight of dry gas and of air per lb. carbon.

Formula A:

$$\text{Dry gas per lb. C} = \frac{14.252 \times 11 + 3.508 \times 8 + 82.240 \times 7}{3(14.252 + 1.584)} = 16.008. \text{ pounds.}$$

Formula B:

$$\text{Air per pound C} = 5.8 \frac{2(14.252 + 3.508) + 1584}{14.252 + 1.584} = 13.589 \text{ pounds.}$$

The error in the last result is 15.414 − 13.589 = 1.825 pounds.

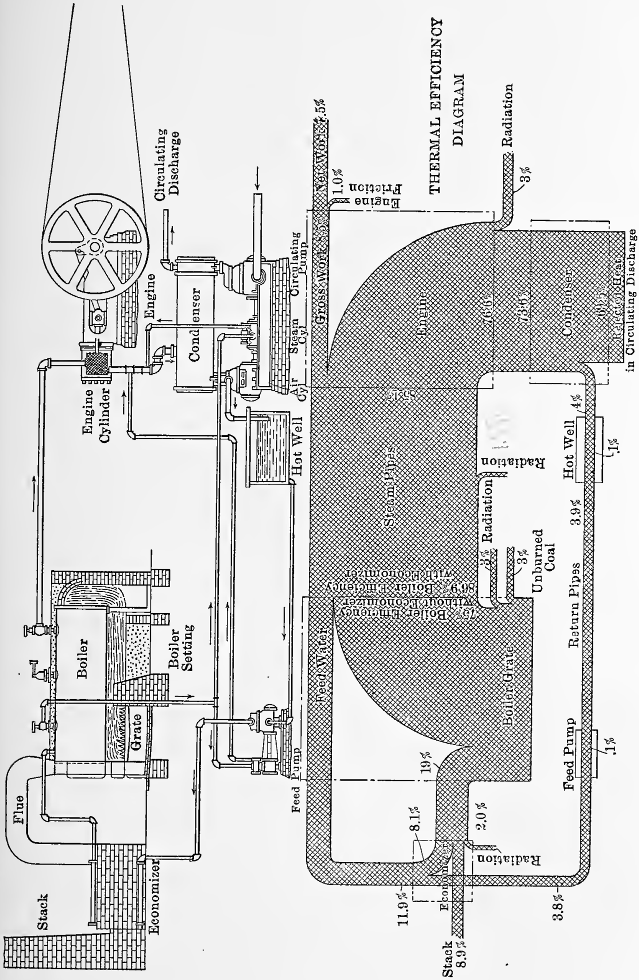


FIG. 180.—Heat Distribution and Losses in a Steam Boiler and Engine Plant.

Professor D. S. Jacobus gives another formula for the air per pound of carbon, in which the error of formula **62** is almost entirely avoided. It is

Formula C:

$$\text{Air per pound C} = \frac{7N}{3(CO_2+CO)} \div 0.77, \text{ or } \frac{N}{0.33(CO_2+CO)}, \qquad (63)$$

in which N, CO_2, and CO are the percentages by volume of these gases. Making the computation from the data of the above analysis, we have:

$$\text{Air per pound C} = \frac{80.656}{0.33(14.252+1.584)} = 15.434 \text{ pounds,}$$

the true value being 15.414 pounds.

Fig. 180 is a diagram showing the heat distribution and losses in a steam boiler and engine plant, due mainly to Parsons. The method of showing graphically the percentage of losses is particularly interesting.

CHAPTER XI

STEAM ENGINE TESTING

Most important of the tests made of nearly all classes of machinery is that for **mechanical efficiency**; meaning the comparison of the useful work performed with the amount of work theoretically possible to obtain with a perfect machine. In other words, in an engine the mechanical efficiency, $\mathbf{E}_m$, is the ratio of the brake horse power to the indicated horse power, or

$$\mathbf{E}_m=\frac{\mathbf{B.H.P.}}{\mathbf{I.H.P.}} \quad \ldots \ldots \quad (64)$$

The difference between the indicated horse power and the brake horse power is called the **friction horse power.** In many cases with very large engines, it is not readily possible to obtain the brake horse power directly, and in such cases it is customary to obtain approximately the horse power lost in friction from a so-called "friction indicator diagram," obtained from the areas of indicator diagrams when the only work done is that required to overcome its own friction, or in common parlance, when the engine is "running light." The brake horse power is then taken to be the difference between the indicated horse power and the friction horse power. Such a determination of friction horse power and of mechanical efficiency by calculation cannot be considered very accurate, because the friction of an engine increases slightly with increasing loads.

Observed and calculated data of mechanical efficiency may be tabulated in the following form:

STEAM ENGINE TESTING

TESTS FOR MECHANICAL EFFICIENCY AND FRICTION

Date....................Test made by..........................
Description of engine tested..
....
Tare of Brake........lbs. Length of brake arm.............feet......
Engine and Brake Constants (see pages 121 and 123.)................

No. of Reading.	Time.	Weight on Brake, lbs.		R. P. M.	Areas of Indicator Cards, sq. ins.		Indicated Horse power.			Brake Horse power.	Friction Horse power.	Mech. Effic. %
		Gross.	Net.		Head End.	Crank End.	Head. End.	Crank End.	Total			

Valve Setting (Slide Valve Engines). In order that steam may be used economically in an engine, it is necessary that the valve be set carefully and accurately, so that when an indicator card is taken the diagram obtained will be as nearly as possible like the ideal. Adjustment of a slide valve on an engine is accomplished in two different ways, with different effects:

(1) By moving the valve on its stem;

(2) By adjusting the eccentric.

To Set the Valve for Equal Leads. The valve should be placed first in such a position on its stem (valve rod) that the amount of its travel will be the same on the two sides of its mid-position. In other words, the valve is to be set on its stem so that its movement will be symmetrical with respect to the ports. Typical slide valves are shown in mid-position in **Figs. 181** and **182.** Setting the valve symmetrically or in mid-position is easily accomplished by turning the engine (or by moving the eccentric on the shaft) until the valve is brought to the farthest point of its travel on one side of its mid-position. Then measure the width [1] of the part of the steam port which is uncovered. The valve should then be shifted to the limit of its travel at the other end so that the width of the uncovered

[1] It is assumed of course that corresponding dimensions of the ports are the same at the two ends of the valve seat.

portion of the steam port can be measured at that end. If the widths measured at the two ends are not the same, the valve must be moved on its stem a distance equal to half the difference between the widths of the uncovered ports. This movement of the valve will be in a direction **away from** the

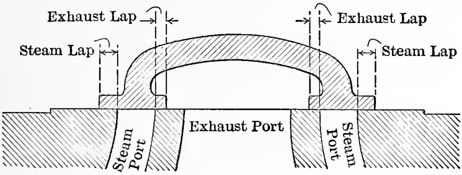


FIG. 181.—Ordinary D-slide Valve in Mid-position.

port having the **smaller** opening. Measurements should be repeated and if the widths of the uncovered ports are still unequal this method of adjustment must be continued.

By the method described the two **leads** of the valve will have been made the same; in other words, the distance the

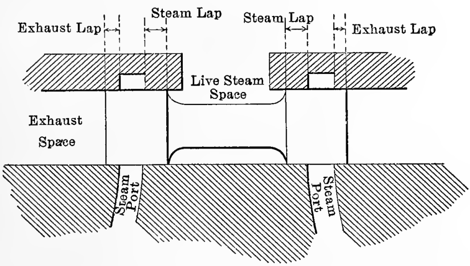


FIG. 182.—Piston Type of Slide Valve in Mid-position.

valve uncovers the steam ports when the engine is on the dead-centers will be the same at both ends of the cylinder. Now move the engine accurately to the dead-center [1] toward the

[1] An engine **can be** put on **dead-center** quite accurately by the "**method of trammels.**" When the engine is just a little off the center to be determined, make small scratch-marks opposite each other both on the cross-head and on one of the guides. Now set a pair of dividers or trammels with one end resting on the bedplate of the engine, its foundation, or some convenient stationary object near the fly-wheel, and with the other end

head end, for example, and move the eccentric on the **shaft in the direction in which the engine is to run**[1] until the steam port at the head end of the cylinder has been opened a distance equal to the lead required and in the position defined in general by stating that a further movement of the valve in the same direction will open the port wider. In this position the eccentric should be firmly fixed to the shaft. The engine should then be turned over to the other dead-center so that the lead can be measured at that end. If the leads are not the same, the difference should be halved, one part to be taken up by moving the valve on its stem, and the other by moving the eccentric.

To set the valve for equal cut-offs, the valve is first set on its stem so that the travel will be the same on both sides of the mid-position as explained at the bottom of page 230 to make its movement symmetrical with the ports. Now an adjustment of the eccentric must be made so that steam will be cut off at each end of the cylinder when the piston has moved the same distances or the same per cent of the stroke from the two ends. To perform this adjustment, mark on the cross-head as accurately as possible the limits of the stroke, and set the cross-head at the per cent of the stroke for cut-off appearing to be most suitable for the conditions of load. Move the eccentric on the shaft in the direction in which the engine is to run until it can be seen that the valve would be just closing the steam port at the end of the cylinder from which the piston is moving. Fasten the eccentric securely in this position and turn the engine over to observe whether the valve will be just closing the other steam port when the piston has moved the same distance, measured on the cross-head,

mark a point on the fly-wheel. The engine should then be moved over or beyond the dead-center until the marks made on the cross-head and on the guide come together again. With the dividers set with the points the same distance apart as before again put a mark on the fly-wheel. Then if the engine is turned back so that the end of the dividers used to mark on the fly-wheel is at a point half way between the two marks, it will be set quite accurately on the dead-center required. In all these adjustments care must be taken to turn the engine each time in the same direction with respect to the dead-center so that the lost motion or back lash is taken up in the same direction.

[1] This applies only to a valve like the one in **Fig. 181**, which takes steam on the "outside." When the valve takes steam on the inside (**Fig. 182**) the eccentric must be moved in the opposite direction.

from the other end of the cylinder. If the setting is not correct, the error should be halved, correcting for one half by moving the valve on the stem and for the other by moving the eccentric on the shaft. This operation, which is a "cut and try" process, must be repeated until the required setting is secured.

Methods similar to those described are preferred usually for the accurate setting of the slide valves of slow- and medium-speed engines. High-speed engines as well as slow-speed engines of the Corliss type have their valves set usually on the basis of the information secured from indicator diagrams taken on the engines, showing approximately the "timing" of the events of the stroke. To set a **slide valve** successfully

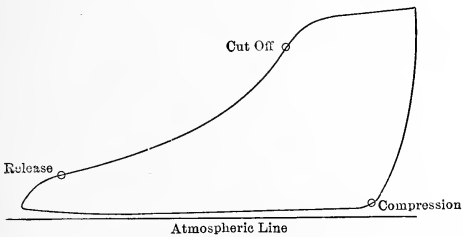


FIG. 183.—Indicator Diagram Illustrating the Point of Cut-off.

by the "indicator" method, the valve and ports should be measured to determine the "lap" dimensions and port openings indicated in **Fig. 181**, page 231, and the valve travel by direct measurements. With these data a **Zeuner**[1] **valve diagram** should be constructed, showing a good steam distribution for assumed lead or cut-off. Then construct the theoretical indicator card from the Zeuner diagram and adjust the setting of the valve on the stem and the eccentric on the shaft till a close approximation to the theoretical card is obtained. In this adjustment the first thing to be done is to equalize the valve travel by locating it on its stem so that the travel will be the same on both sides of its mid-position.

[1] It is beyond the scope of this book to take up a discussion of valve diagrams. The theory and construction of the Zeuner diagram is given in nearly all books on the steam engine.

Use a spring light enough to give an indicator diagram about $1\frac{1}{2}$ inches high so that events of the stroke,—admission, cut-off, release, and compression, will be shown as clearly as possible. Sometimes it is difficult to determine these events on a diagram on account of the curves gradually running into each other without the point separating the different curves being clearly defined. A good method for such cases is to produce along their regular trend both of the curves of which the intersection is required and take for the intersection the point where these curves cross each other. The method is illustrated on an indicator card in **Fig. 183** showing the point of cut-off.

In a slide valve engine it is not possible to set the valve to secure at the same time equal cut-offs and equal leads.

Ideal and imperfect indieator diagrams taken from slide valve and Corliss engines are shown in **Fig. 184.** A little study of such diagrams may help to solve many difficulties in valve setting.

Setting Corliss Valves. A brief desciiption[1] of the essential parts of the valve gear of a Corliss engine will assist in obtaining a clearer conception of the subject. In **Figs. 185** and **186** similar capital letters of reference indicate the same parts of the mechanism.

Fig. 185 shows all the essential parts of the valve gear. The steam valves work in the chambers **S, S** and the exhaust valves work in the chambers **E, E.** The double-armed levers **AC, AC,** work loosely on the hubs of the valve-stem brackets and the lever arms **B, B;** the former are connected to the wrist plate **W** by the rods **M, M;** the levers **B, B** are keyed to the valve stems **V, V,** and are also connected by the rods **O, O** to the dashpots **D, D.** The double-armed levers carry at their outer ends, **C, C** hardened steel catch plates, which engage with arms **B, B,** making the two arms **B** and **C** work in unison until steam is to be cut off; at this point another set of levers **H, H,** connected by the cam rods **G, G** to the governor, come into play, causing the catch plates to release the arms **B, B,** the outer ends of which are then pulled downward by the weight of the dashpot plunger, causing the steam valves

[1] This description is from *American Machinist,* vol. 18, page 391. For clearness the article is considered unusually good.

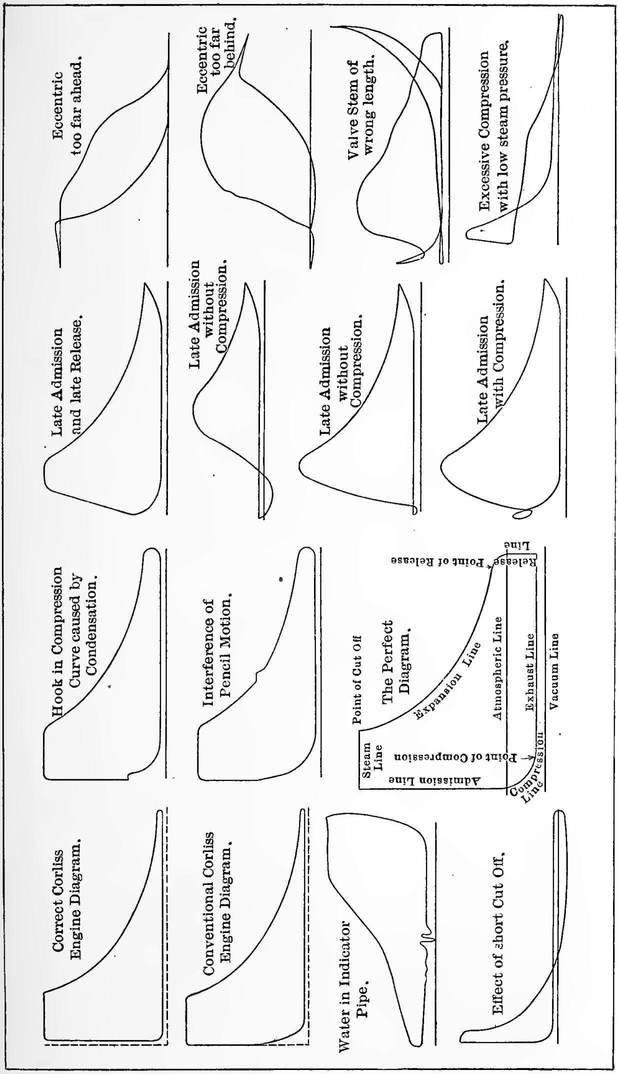


FIG. 184.—Samples of Ideal and Imperfect Indicator Diagrams.

to rotate on their axes and thus cut off steam. These are the essential features of the Corliss gear, although the design of the mechanism is greatly modified by different builders.

The exhaust valve arms **F** are connected to the wrist plate by the rods **N, N,** and it is seen that all the valves receive their motion from the wrist plate; the latter receives its motion from the hook rod **I**; this rod is generally attached to a rocker arm, not shown; to this arm the eccentric rod is

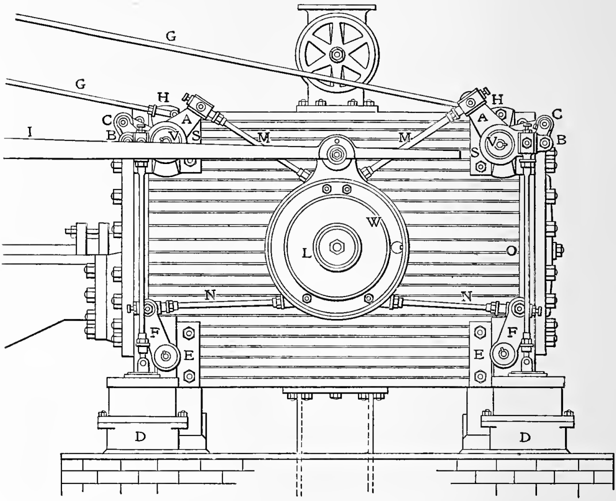


Fig. 185.—Corliss Valve Gear.

also attached. The carrier arm is usually placed about midway between the wrist plate and eccentric, and in the center of its travel stands in a vertical position.

The setting of the valves is not a difficult matter when, on the wrist plate, its support, valves and cylinder, the customary marks have been placed for finding the relative positions of wrist plate and valves.

Now referring to **Fig. 186,** when the back bonnets of the valve chambers have been taken off, there will be found a mark or line **a** on the end of each steam valve s, s, coinciding

with the working or opening edges of each valve; another line **b** will be found on each face of the steam valve chamber coinciding with the working edge of the valve, and the line **h,** on the face of each exhaust valve chamber, coincides with the working edge of the exhaust port. On the hub of the wrist plate will be found a line **d,** coinciding with the center line **d, k;** lastly, there are three lines **f, c, f,** on the hub of the wrist-plate support, placed in such a way that when the line **d** coincides with the line **c,** the wrist plate will stand exactly in the center of its motion, and when the line **d** coincides with either of the lines **f, f,** the wrist plate will be at one of the extreme ends **u** or **v** of its travel. It should be noticed that since the lines **f, e, f** are drawn on the periphery of the hub of

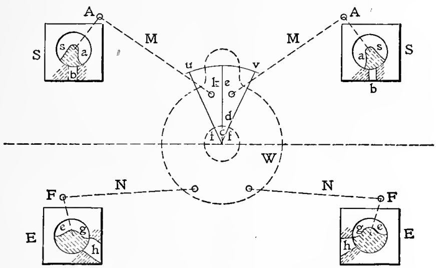


Fig. 186.—Diagram of a Corliss Valve Mechanism.

the wrist-plate support, and the line **d** is drawn on the periphery of the wrist-plate hub, these lines cannot stand in a vertical line, as shown; we have adopted this way of showing them, simply for the purpose of making the matter plain.

In setting the valves, the first step will be to set the wrist plate in its central position, so that lines **c** and **d** will coincide, and fasten the wrist plate in this position by placing a piece of paper between it and the washer **L** on its supporting pin. Now set the steam valves so that they will have a slight amount of lap, that is to say, the lines **a, a** must have moved a little beyond the lines **b, b;** the amount of this lap depends much on individual preference and experience; it ranges from $\frac{1}{16}$ to $\frac{1}{4}$ inch for small engines, and from $\frac{1}{4}$ to $\frac{9}{16}$ inch for comparatively large engines. This lap is obtained by length-

ening or shortening the rods **M, M** by means of the adjusting nuts.

Now place the exhaust valves **e, e,** by lengthening or shortening the rods **N, N** by means of the adjusting nuts in a position so that the working edges will just open the exhaust ports, or, in other words, place the lines **g** and **h** nearly in line with each other. Some engineers prefer a slight amount of lap, others prefer a slight opening of the exhaust ports when the valves are in this position; under these conditions the lines **g** and **h** cannot be in line, but will stand apart, as indicated in the illustration; the distance between these lines will, of course, be equal to the desired amount of opening; for small engines it is about $\frac{1}{16}$ inch, and for larger engines may be increased to $\frac{3}{16}$ inch, but in any case the amount of this opening should be less than the lap of the steam valves, otherwise there will be danger of steam blowing through.

The paper between the wrist plate and the washer on the supporting pin should now be taken out, so that the wrist plate connected to the valves can be swung on its pin.

The next step will be to pay some attention to the rocker arm. Set this arm in a vertical position by means of a plumb-line, and connect the eccentric rod to it; then turn the eccentric around on the shaft, and see that the extreme points of travel are at equal distances from the plumb-line. To secure this a little adjustment in the stub end of the eccentric rod may be necessary. Now connect the hook rod **I** to its pin on the wrist plate, and again turn the eccentric around on the shaft, and thus determine the extreme points of travel of the wrist plate. If all parts have been correctly adjusted, the line **d** will coincide with the lines **f, f** at the extreme points of travel; if this is not the case, the hook rod will have to be adjusted at its stub end so as to obtain the desired equalized motion of the wrist plate.

The next step will be to set the valves correctly with the position of the crank; to do so the lengths of the rods **M, M, N,** and **N** must not be changed, but the following mode of procedure should be followed: Place the crank on one of its dead-centers, and turn the eccentric loosely on the shaft in the direction in which the engine is to run, until the steam valve nearest to the piston shows an opening or lead of $\frac{1}{32}$ to $\frac{1}{8}$ inch,

according to size of engine, the smaller lead, of course, being adopted for small engines. After the proper lead has been given to this valve, secure the eccentric, and turn the shaft with eccentric in the same direction in which the engine is to run until the crank is on the opposite dead-center, and notice if the opening or lead at this end of the cylinder is the same as on the other steam valve; if not, shorten or lengthen slightly, as may appear necessary, the connection between wrist plate and eccentric; of course much adjustment in the length of these connections is not admissible witheout rsetting the valves with reference to the wrist plate.

The only thing which now remains to be done is to adjust the cam rods **G, G.** To do so, secure the governor balls in their highest position, and disconnect the hook rod from wrist pin; lengthen or shorten the cam rods **G, G,** so as to bring the detachment apparatus into action, swing the wrist plate back and forward and make such adjustment in the rods **G, G,** as to permit the steam valves to be released when the steam port has been opened about $\frac{1}{8}$ inch. This adjustment is for the purpose of keeping the engine under the control of the governor, in case, for some reason or another, the load on the engine is suddenly thrown off. After this adjustment the governor balls should be placed in their lowest position, in which the releasing gear should not detach the steam valves, but allow the steam to follow nearly full stroke. Sometimes the releasing gear is constructed in such a manner as to close the steam valves automatically, in case the belt leading to the governor should be broken, or the load on the engine suddenly thrown off. In cases of this kind the governor balls need not be placed in their highest position, but should be placed in their lowest position, and the wrist plate moved to either end of its extreme travel; the steam port opposite this end of travel of wrist plate will then·be wide open; now adjust the corresponding cam rod so that the releasing gear is nearly on the point of releasing the valve; then move the wrist plate to other end of its extreme travel, and adjust the other cam rod in the same manner. To prove the correctness of the cut-off adjustment, raise the governor balls to about a position where they would be when at work, or to a medium height, and block them there; then, with the connection made

between the eccentric and the wrist plate, turn the engine shaft slowly in the direction in which it is to run, and when the valve is released measure upon the slide the distance through which the cross-head has moved from its extreme position. Continue to turn the shaft in the same direction, and when the other valve is released, measure the distance through which the cross-head has moved from its extreme position, and if the cut-off is equalized, these two distances will be equal to each other. If they are not, adjust the length of the cam rods until the points of cut-off are at equal distances from the beginning of the stroke. Replace the back bonnets and see that all connections have been properly made, which will complete the setting of the valves. Wherever convenient, it is desirable that an indicator be applied to the engine when at work, and the setting of the valves tested. If necessary, they should be readjusted for the best possible condition for economical work.

Clearance Determination of an Engine. This test is made usually to determine the clearance volume of a steam or a gas engine. It is sometimes important to know the clearance volume of an engine, as it materially affects the expansion curve of the engine. If it is too large it causes an excessive loss in the engine. The clearance volume is also necessary if a theoretical expansion curve is to be constructed.

The engine is first set on dead-center with the piston at the head end of the cylinder. This is done by the "method of trammels" (see footnote, page 231). Then the steam chest cover and valve are to be removed and a rubber gasket under a block of wood is placed over both steam-port openings in the valve seat and bolted on. Usually candle wicking must be packed around the piston to stop excessive leakage. For this purpose the cylinder head must be removed and again replaced.

Two vessels filled with clean water should be provided and weighed. The clearance space is to be filled from one vessel, the time required being taken. As soon as the space is filled the first vessel is removed and the space is kept filled with water from the other vessel for five minutes. The vessels are then again weighed and the water used from each of them determined. The average rate of leakage while filling the space is

usually assumed to be one-half the rate of leakage when full of water as during the leakage test.

If w_1 = weight in pounds to fill the clearance space, t minutes = the time required to fill the clearance space, and w_2 = the weight of water in pounds necessary to keep it full for one minute; then the leakage during filling is

$$w' = \frac{w_2}{2} \times t,$$

and the clearance = $(w_1 - w')$ in pounds of water, which can be readily reduced to cubic inches.

The clearance for the crank end is found in the same general way as that for the head end.

Most engines have small holes at the top of the cylinder at each end (for double-acting engines) which lead into the clearance space. Holes which are covered by the piston on the dead-center would obviously be of no value. All water must of course be drained from each end before filling. Removing the cylinder head for packing the piston is the best method for observing with certainty that there is no water in the head end. The drip pipes in the cylinder can usually be relied on to remove water from the crank end. Determinations should be repeated several times, and the average value is to be used in calculations.

RULES FOR CONDUCTING STEAM ENGINE TESTS

An elaborate report has been published by the American Society of Mechanical Engineers entitled **"Rules for Conducting Steam Engine Tests."**[1] In this report the rules regulating standard commercial tests of engine plants, including auxiliary machinery, are given in great detail. Every engineer should have available a copy of these rules. It is impracticable to print in this book the complete report, but space will be taken to give the most important parts.

[1] This report is issued in a pamphlet of 78 pages and can be obtained at at small cost from the Secretary of the American Society of Mechanical Engineers, Engineering Society's Building, New York city. It is printed complete in the Transactions of the Society, Vol. 24, pages 713–846.

One of the most important subjects discussed is in regard to the definition of the "Heat consumption" of an engine plant. This is to be determined by measuring the quantity of steam consumed by the plant, calculating the total heat of the entire quantity and crediting this total with that portion of the heat rejected by the plant which is utilized and returned to the boiler. "Engine plant" as used in this report should include the entire equipment of the steam plant producing power; that is, the main cylinder or cylinders, the steam jackets and reheaters, the air, circulating and boiler-feed pumps if steam driven, and any other machinery driven by steam required for the operation of the engine. That the engine plant should be charged with the steam used by all the auxiliary machinery in determining the plant economy is necessary because the steam consumption of the engine is finally benefited, or at least it should be, by the heat they return to the system. It is, of course, now the general practice in commercially operated plants to pass the exhaust steam from auxiliaries operated non-condensing, through a feed-water heater, thus carrying back to the boiler a great deal of the heat.

When a plant is **operating non-condensing,** discharging the steam into the atmosphere, **or with a jet condenser,** the **steam consumption of the engine cannot be determined by weighing or measuring the steam used** as can be done when a surface condenser is used. The method followed in this case is to determine the steam used by the weight of water supplied to the boiler, assuming, of course, that all the steam from the boiler or boilers used goes to the engine tested. It can usually be arranged for a test of one of the engines in a large plant that one or more boilers can be segregated or cut off in the piping connections so that these boilers alone supply the engine. When, however, this method is to be used it is necessary to determine by a separate test the **leakage of the boiler and of the piping** from the boiler to the throttle valve of the engine. This leakage, of course, is the amount of feed-water pumped into the boiler to keep the level in the water gage constant without taking away any more steam than is lost in this way. When determining this leakage, the pressure in the boiler must be maintained the same as that at which steam is to be supplied to the engine for its test. The feed-water pumped

into the boiler supplying the engine less the boiler and pipe leakage will be the net amount of steam used by the engine.

Surface condensers are usually perferred for accurate engine tests because the steam used by the engine can be determined directly by weighing or measuring the condensed steam. A surface condenser is essentially a vessel of considerable size in which there are a great many brass tubes. It is usually designed so that exhaust steam passes on its way through the condenser into contact with the outside surface of these tubes while cold water for condensing the steam circulates inside the tubes. Steam condensed in this way accumulates in the bottom of the condenser and is removed by an air pump used for producing a vacuum, or by gravity if the pressure in the condenser is atmospheric, as it will be if the engine is operating "non-condensing," that is, without a vacuum. It is very essential, however, that **surface condensers be tested for leakage,** preferably before and after every important test is made; and the leakage should be determined with the same vacuum in the condenser as there is when it is used in the engine test. Probably the best method to determine the amount of condenser leakage is to pass cooling water through the tubes at the normal rate of flow, maintaining at the same time with the air pump the required vacuum. Then the amount of water removed by the air pump is the leakage of circulating water through the tubes into the steam space. Under normal operating conditions there would be **no leakage of steam into the circulating water,** because the water will be at higher pressure than the steam.

If only a test is to be made to determine whether or not there is any leakage in the condenser, the test most generally applied is to close all connections to the condenser and observe whether a vacuum once established can be maintained a reasonably long time. A more rapid test, applicable, however, only where salt water is used for cooling, is to test the condensed steam several times during a test by adding a little silver nitrate to a small amount of the condensed steam. If there is no appreciable precipitate [1] it may be assumed that the condenser does not leak.

Some of the important considerations to be observed in an

[1] The white precipitate formed with the salt in sea water is of course silver chloride, thus, $AgNO_3 + NaCl = AgCl + NaNO_3$.

accurate engine test will now be given as stated in the rules adopted by the American Society of Mechanical Engineers.

General Conditions of Plant. Examine the engine and the entire system of piping concerned in the test; note its general condition and any points of design, construction, or operation which bear on the objects in view. Make a special examination of the valves and pistons for leakage by applying the working pressures with the engine at rest, and observe the quantity of steam, if any, blowing through per hour.

Dimensions, etc. Measure the dimensions of the cylinders of the engine when it is hot. If they are much worn, the **average** diameter should be determined. Measure also the **clearance,** which should be done, if possible, by filling the spaces with water previously measured, the piston being placed at the end of the stroke (see page 240). If the clearance cannot be measured directly, it can be determined approximately from the working drawings of the cylinder.

Calibration of Instruments. All instruments and apparatus should be calibrated and their reliability and accuracy verified by comparison with recognized standards.

Leakages of Steam, Water, etc. In all tests except those of a complete plant made under conditions as they exist, the boiler and its connections, both steam and feed, as also the steam piping leading to the engine and its connections, should, so far as possible, be made tight. All connections should, so far as possible, be **visible** and be **blanked off,** and where this cannot be done, satisfactory assurance should be obtained that there is no leakage either in or out.

Duration of Tests. The duration of a test should depend largely upon its character and the objects in view. The standard heat test of an engine, and, likewise, a test for the simple determination of the feed-water consumption, should be continued for at least five hours, unless the class of service precludes a continuous run of so long duration. It is desirable to continue the test, the number of hours stated to obtain a number of consecutive hourly records as a guide in analyzing the reliability of the whole.

The commercial test of a complete plant, embracing boilers as well as engine, should continue at least one full day of twenty-four hours, whether the engine is in motion during the entire

time or not. A continuous coal test of a boiler and engine should be of at least ten hours' duration, or the nearest multiple of the interval between times of cleaning fires.

Starting and Stopping a Test. (a). **Standard Heat Test and Feed-water Test of Engine.** The engine having been brought to the normal condition of running, and operated a sufficient length of time to be thoroughly heated in all its parts, and the measuring apparatus having been adjusted and set to work, the height of water in the gage glasses of the boilers is observed, the depth of water in the reservoir from which the feed water is supplied is noted, the exact time of day is observed, and the test held to commence. Thereafter the measurements determined upon for the test are begun and carried forward until its close. If practicable, the test may be commenced at some even hour or minute, but it is of first importance to begin at such time as reliable observations of the water heights are obtained, whatever the exact time happens to be when these are satisfactorily determined. When the time for the close of the test arrives, the water should, if possible, be brought to the same height in the glasses and to the same depth in the feed-water reservoir as at the beginning, delaying the conclusion of the test, if necessary, to bring about this similarity of conditions. If differences occur, the proper corrections must be made.

(b). **Complete Engine and Boiler Tests.** For a continuous running test of combined engine or engines, boiler or boilers, the same directions apply for beginning and ending the feed-water measurements as those just referred to. The time of beginning and ending such a test should be the regular time of cleaning the fires, and the exact time of beginning and ending should be the time when the fires are fully cleaned, just preparatory to putting on fresh coal.

For a commercial test of combined engine and boiler, whether the engine runs continuously for the full twenty-four hours of the day, or only a portion of the day, the fires in the boilers being banked during the time when the engine is not in motion, the beginning and ending of the test should occur at the regular time of cleaning the fires, the method followed being that given above. In cases where the engine is not in continuous motion, as, for example, in textile mills, where the working time is ten or eleven hours out of twenty-four, and the fires are cleaned

and banked at the close of the day's work, the best time for starting and stopping a test is the time just before banking, when the fires are well burned down and the thickness and condition can be most satisfactorily judged.

Measurements of Heat Units Consumed by the Engine. The measurements of the heat consumption require the measurement of each supply of feed-water to the boiler—that is the water supplied by the main feed pump, that supplied by auxiliary pumps, such as jacket water, water from separators, drips, etc., and the water supplied by gravity and other means; also the determination of the temperature of the water supplied from each source, together with the pressure and quality of the steam. The temperatures at the various points should be those applying to the working conditions.

The heat to be determined is that used by the entire engine equipment, embracing the main cylinders and all auxiliary cylinders and mechanism concerned in the operation of the engine, including the air pump, circulating pump, feed pumps, also the jacket and reheater, when these are used.

The steam pressure and the quality of the steam are to be taken at some point conveniently near the throttle valve. The quantity of the **steam used by the calorimeter must be determined and properly allowed for.**

Measurement of Feed-water or Steam Consumption of Engine, etc. The method of determining the steam consumption applicable to all plants is to measure all the feed-water supplied to the boilers, and deduct therefrom the water discharged by separators and drips, as also the water and steam which escapes on account of leakage of the boiler and its pipe connections and leakage of the steam main and branches connecting the boiler and the engine. In plants where the engine exhausts into a surface condenser the steam consumption can be measured by determining the quantity of water discharged by the air pump, corrected for any leakage of the condenser, and adding thereto the steam used by jackets, reheaters, and auxiliaries as determined independently.

The corrections or deductions to be made for leakage above referred to should be applied only to the standard heat-unit test and tests for determining simply the steam or feed-water consumption, and not to coal tests of combined engine and

boiler equipment. In the latter, no correction should be made except for leakage of valves connecting to other engines and boilers, or for steam used for purposes other than the operation of the plant under test. Losses of heat due to imperfections of the plant should be charged to the plant, and only such losses as are concerned in the working of the engine alone should be charged to the engine.

Measurement of Steam Used by Auxiliaries. It is highly desirable that the quantity of steam used by the auxiliaries, and in many cases that used by each auxiliary, should be determined exactly, so that the net consumption of the main engine cylinders may be ascertained and a complete analysis made of the entire work of the engine plant.

Indicated Horse Power. The indicated horse power should be determined from the average mean effective pressure of diagrams taken at intervals of twenty minutes, and at more frequent intervals if the nature of the test makes it necessary, for each end of the cylinder. With variable loads, such as those of engines driving generators for electric railroad work, and of rolling-mill engines, the diagrams cannot be taken too often.

The most satisfactory driving rig for indicating seems to be some form of well-made pantagraph, with driving cord of fine annealed wire leading to the indicator. The reducing motion, whatever it may be, and the connections to the indicator, should be so perfect as to produce diagrams of equal lengths, when the same indicator is attached to either end of the cylinder, and produce a proportionate reduction of the motion of the piston at every point of the stroke, as proved by test.

The use of the three-way cock and the single indicator connected to the two ends of the cylinder is not advised, except in cases where it is impracticable to use an indicator close to each end. If a three-way cock is used, the error produced by the increased clearance should be determined and allowed for.

Testing Indicator Springs. To make a perfectly satisfactory comparison of indicator springs with standards, the calibrations should be made, if practicable, under the same conditions as those pertaining to their ordinary use.

Brake Horse Power. This term applies to the power delivered from the fly-wheel shaft of the engine. It is the power

absorbed by a friction brake applied to the rim of the wheel, or to the shaft. A form of brake is preferred that is self adjusting to a certain extent, so that it will, of itself, tend to maintain a constant resistance at the rim of the wheel. One of the simplest brakes for comparatively small engines, which may be made to embody this principle, consists of a cotton or hemp rope, or a number of ropes, encircling the wheel arranged with weighing scales or other means of showing the strain. An ordinary band brake may also be constructed so as to embody the principle. The wheel should be provided with interior flanges for holding water used for keeping the rim cool.

Quality of Steam. When saturated steam is used, its quality should be obtained by the use of a good throttling calorimeter attached to the main steam pipe near the throttle valve. When the steam is superheated, the amount of superheating should be found by the use of a thermometer placed in a thermometer-well filled with mercury, inserted in the pipe. The sampling pipe for the calorimeter should, if possible, be attached to a section of the main pipe having a vertical direction, with the steam preferably passing upward, and the sampling nozzle should be made of a half-inch pipe, having at least twenty one-eighth inch holes in its perforated surface.

Speed. There are several reliable methods of ascertaining the speed, or the number of revolutions of the engine crankshaft per minute. The most reliable method, and the one recommended, is the use of a continuous recording engine register or counter, taking the total reading each time that the general test data are recorded, and computing the revolutions per minute corresponding to the difference in the readings of the instrument. When the speed is above 250 revolutions per minute, it is almost impossible to make a satisfactory counting of the revolutions without the use of some form of mechanical counter.

Recording the Data. Take note of every event connected with the progress of the trial, whether it seems at the time to be important or unimportant. Record the time of every event, and the time of taking every weight, and every observation. Observe the pressures, temperatures, water heights, speeds, etc., every twenty or thirty minutes when the conditions are practically uniform, and at much more frequent intervals if the conditions vary.

Uniformity of Conditions. In a test having for its object the determination of the maximum economy obtainable from an engine, or where it is desired to ascertain with special accuracy the effect of predetermined conditions of operations, it is important that all the conditions under which the engine is operated should be maintained uniformly constant.

Analysis of Indicator Diagrams. (a) Steam Accounted for by the Indicator. The simplest method of computing the steam accounted for by the indicator is the use of the formula

$$\mathbf{M} = \frac{13750}{\mathbf{M.E.P.}}[(\mathbf{C}+\mathbf{E})\times\mathbf{W}_c - (\mathbf{H}+\mathbf{E})\times\mathbf{W}_h], \quad . \quad . \quad (65)$$

which gives the weight in pounds per indicated horse power per hour. In this formula the symbol **M.E.P.** refers to the mean effective pressure. In multiple-expansion engines this is the "combined" mean effective pressure referred to the cylinder in question. **C** is the proportion of the stroke completed at a point on the expansion line of the diagram near the actual cut-off or release, the symbol **H** to the proportion of compression, and the symbol **E** to the proportion of clearance; all of which are determined from the indicator diagram. The symbol $\mathbf{W}_c$ refers to the weight of one cubic foot of steam at the cut-off or release pressure; and $\mathbf{W}_h$ to the weight of one cubic foot of steam at the compression pressure, these weights being taken from steam tables.

The points at cut-off and release on the expansion line and the point at the beginning of compression are located as shown on the sample diagram, **Fig. 187.** On the expansion and compression lines they are the points which mark the complete closure of the valve. The point at cut-off lies where the curve of expansion begins after the rounding of the diagram due to throttling or "wire-drawing" which occurs when the valve is closing. This point of cut-off may be located on the diagram by finding the point where the curve is tangent to a hyperbolic curve.

Should the point selected in the compression curve be at the same height as the point in the expansion curve, then $\mathbf{W}_c = \mathbf{W}_h$, and the formula becomes

$$\frac{13750}{\mathbf{M.E.P.}}\times(\mathbf{C}-\mathbf{H})\times\mathbf{W}_c, \quad . \quad . \quad . \quad . \quad (66)$$

in which (**C**—**H**) represents the distance between the two points divided by the length of the diagram.

When the load and all other conditions are substantially uniform, it is unnecessary to work up the steam accounted for by the indicator from all the diagrams taken. Five or more sample diagrams may be selected and the computations based on the samples instead of on the whole.

(b) **Sample Indicator Diagrams.** In order that the report of a test may afford complete information regarding the conditions of the test, sample indicator diagrams should be selected from those taken and copies appended to the tables of results. In cases where the engine is of the multiple-expansion type

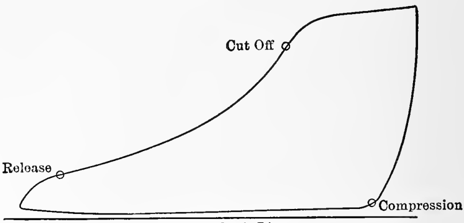


FIG. 187.—Typical Indicator Diagram.

these sample diagrams may also be arranged in the form of "combined" diagrams.

(c) **The Point of Cut-off.** The term "cut-off" as applied to steam engines, although somewhat indefinite, is usually considered to be at an earlier point in the stroke than the beginning of the real expansion line. That the cut-off point may be defined in exact terms for commercial purposes, as used in steam-engine specifications and contracts, the Committee recommends that, unless otherwise specified, the **commercial cut-off,** which seems to be an appropriate expression for this term, be ascertained as follows: Through a point showing the maximum pressure during admission, draw a line parallel to the atmospheric line. Through the point on the expansion line near the actual cut-off, draw a hyperbolic curve. The point where these two lines intersect is to be considered the point at the commercial cut-off. The percentage is then found by dividing the

length of the diagram measured to this point, by the total length of the diagram, and multiplying the result by 100.

The principle involved in finding the commercial cut-off is shown in **Figs. 188** and **189**. The first represents a diagram

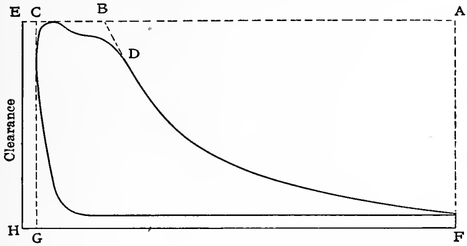


FIG. 188.—Indicator Diagram from a Slow-speed Corliss Engine.

from a slow-speed Corliss engine and the second a diagram from a single-valve high-speed engine. In the latter case, owing to the inertia or "fling" of the indicator pencil, the steam line is irregular and the maximum pressure is found by taking the mean of the vibrations at the highest part of the curve.

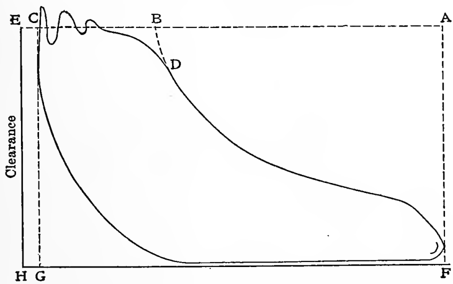


FIG. 189.—Indicator Diagram from a High-speed Single-valve Engine.

The commercial cut-off, as thus determined, is situated at an earlier point of the stroke than the actual cut-off referred to in computing the "steam accounted for" by the indicator on page 249.

(d) **Ratio of Expansion.** The **commercial** ratio of expansion for a **simple engine** is determined by dividing the volume corresponding to the piston displacement, including clearance, by the volume of the steam at the commercial cut-off, including clearance.

In the **multiple expansion engine** it is determined by dividing the net volume of the steam indicated by the low pressure diagram at the end of the expansion line, assumed to be continued to the end of the stroke by the net volume of the steam at the maximum pressure during admission to the high pressure cylinder.

The **ideal** ratio of expansion is the quotient obtained by dividing the volume of the piston displacement by the volume of the steam at the cut-off (the latter being referred to the throttle-valve pressure) less the volume equivalent to that retained at compression. In a multiple expansion engine the volumes to be used are those pertaining to the low pressure cylinder and the high pressure cylinder, respectively.

(e) **Diagram Factor.** The diagram factor is the proportion borne by the actual mean effective pressure measured from the indicator diagram to that of a diagram in which the various operations of admission, expansion, release and compression are carried on under assumed conditions. The factor recommended refers to an ideal diagram which represents the maximum power obtainable from the steam accounted for by the indicator diagrams at the point of cut-off, assuming first that the engine has no clearance; second, that there are no losses through wire-drawing the steam either during admission or release; third, that the expansion line is a hyperbolic curve; and fourth, that the initial pressure is that of the boiler and the back pressure that of the atmosphere for a non-condensing engine, and of the condenser for a condensing engine.

In cases where there is a considerable loss of pressure between the boiler and the engine, as where steam is transmitted from a central plant to a number of consumers, the pressure of the steam in the supply main should be used in place of the boiler pressure in constructing the diagrams.

Standards of Economy and Efficiency. The hourly consumption of heat, determined by employing the actual temperature of the feed-water to the boiler, as pointed out in the

section entitled "Measurement of Heat Units Consumed by the Engine" on page 246, divided by the indicated and brake horse power, that is, the number of heat units consumed per indicated and per brake horse power per hour, are the standards of engine efficiency recommended by the Committee. The consumption per hour is chosen rather than the consumption per minute so as to conform with the designation of time applied to the more familiar units of coal and water measurement which have heretofore been used. The British standard where the temperature of the feed-water is taken as that corresponding to the temperature of the back-pressure steam, allowance being made for any drips from jackets or reheaters, is also included in the tables.

It is useful in this connection to express the efficiency in its more scientific form, or what is called the **"thermal efficiency ratio."** The thermal efficiency ratio is the proportion which the heat equivalent of the power developed bears to the total amount of heat actually consumed, as determined by test. The heat converted into work represented by one horse power is 1,980,000 foot-pounds per hour, and this divided by 778 equals 2545 B.T.U. Consequently, the thermal efficiency ratio is expressed by the fraction

$$\frac{2545}{\text{B.T.U. per I.H.P. per hour}}.$$

Heat Analysis. For certain scientific investigations it is useful to make a heat analysis of the indicator diagram, to show the interchange of heat from steam to cylinder walls, etc., which is going on within the cylinder. This is unnecessary for commercial tests.

Entropy-temperature Diagram. The study of the heat analysis is facilitated by the use of the entropy-temperature diagram in which areas represent quantities of heat, the co-ordinates being the entropy and absolute temperature.

Ratio of Economy of an Engine to that of an Ideal Engine. The ideal engine recommended for obtaining this ratio is that which was adopted by the Committee appointed by the *Civil Engineers* of London, to consider and report a standard thermal efficiency for steam engines. This engine is one which follows the Rankine cycle where steam at a constant pressure is admitted

into a cylinder having no clearance, and after the point of cut-off is expanded adiabatically to the back-pressure. In obtaining the economy of this engine the **feed-water is assumed to be returned to the boller at the exhaust temperature** (page 261).
returned to the boiler at the exhaust temperature.

The ratio of the economy of an engine to that of the ideal engine is obtained by dividing the heat consumption per indicated horse power per minute for the ideal engine by that of the actual engine.

Miscellaneous. In the case of the tests of combined engines and boiler plants, where the full data of the boiler performance are to be determined, reference should be made to the directions given by the Boiler Test Committee of the Society, Code 1899. (See pages 203–228.)

In testing steam pumping engines and locomotives in accordance with the standard methods of conducting such tests, recommended by the Committees of the Society, reference should be made to the reports of those Committees in the *Transactions American Society of Mechanical Engineers,* vol. 12, page 530, and in vol. 14, page 1312.

Report of Tests. The data and results of the test should be reported in the manner and in the order outlined in one of of the following tables, the first of which gives, it is hoped, a complete summary of all the data and results as applied not only to the standard heat-unit test, but also to tests of a combined engine and boiler for determining all questions of performance, whatever the class of service.

It is recommended that any report of a test be supplemented by a chart in which the data of the tests are graphically presented. (As example of such a chart as applied to a boiler test see page 202.)

DATA AND RESULTS OF STANDARD HEAT TEST OF STEAM ENGINE

Arranged according to the Short Form Advised by the Engine Test Committee of the American Society of Mechanical Engineers. Code of 1902.

1. Made by .... on Engine located at ....
....
Test made to determine ....
2. Date of Trial ....

3. Type and class of engine; also of condenser..........................
4. Dimensions of main engine:
   (a) Diameter of cylinder, inches..............................
   (b) Stroke of piston, feet ....
   (c) Diameter of piston rod, inches ....
   (d) Average clearance, per cent ....
   (e) Horse power constant for one pound mean effective pressure and one revolution per minute...........................
5. Dimensions and type of auxilaries..............................

TOTAL QUANTITIES, TIME, ETC.

6. Duration of test, hours......................................
7. Total water fed to boilers from main source of supply, lbs..........
8. Total water fed from auxiliary supplies:
   (a).. " ....
   (b).. " ....
   (c).. " ....
9. Total water fed to boilers from all sources............ " ....
10. Moisture in steam or superheating near throttle....% or deg.......
11. Factor of correction for quality of steam........................
12. Total dry steam consumed for all purposes........... lbs..........

HOURLY QUANTITIES

13. Water fed from main source of supply per hour, lbs...............
14. Water fed from auxiliary supplies: " " " ....
   (a).. " ....
   (b).. " ....
   (c).. " ....
15. Total water fed to boilers per hour.................. " ....
16. Total dry steam consumed per hour................. " ....
17. Loss of steam and water per hour due to drips from main steam pipes and to leakage of plant................. lbs..........
18. Net dry steam consumed per hour by engine and auxiliaries, lbs......

PRESSURES AND TEMPERATURES (CORRECTED)

19. Pressure in steam pipes near throttle by gage .....lbs. per sq. in.....
20. Barometric pressure of atmosphere in inches of mercury, ins.........
21. Pressure in receivers by gage ....lbs. per sq. in.....
22. Vacuum in condenser in inches of mercury............. ins.........
23. Pressure in jackets and reheaters by gage ....lbs. per sq. in.....
24. Temperature of main supply of feed-water............deg. F.......
25. Temperature of auxiliary supplies of feed-water:
   (a).. " ....
   (b).. " ....
   (c).. " ....
26. Ideal feed-water temperature corresponding to the pressure of steam in the exhaust pipe, allowance being made for the heat derived from the jacket or reheater drips..............deg. F.......

DATA RELATING TO HEAT MEASUREMENTS

27. Heat units per pound of feed-water, main supply...... B.T.U........
28. Heat units per pound of feed-water, auxiliary supplies:
    (a).. " ....
    (b).. " ....
    (c).. " ....
29. Heat units consumed per hour, main supply........... " ....
30. Heat units consumed per hour, auxiliary supplies:
    (a).. " ....
    (b).. " ....
    (c).. " ....
31. Total heat units consumed per hour for all purposes..... " ....
32. Loss of heat per hour due to leakage of plant, drips, etc. " ....
33. Net heat units consumed per hour:
    (a) By engine alone.............................. " ....
    (b) By auxiliaries............................... " ....
34. Heat units consumed per hour by engine alone, reckoned from temperature given in item 26.................B.T.U........

INDICATOR DIAGRAMS

35. Commercial cut-off in per cent of stroke ....
36. Initial pressure lbs. per sq. in. above atmosphere.................
37. Back pressure at mid stroke above or below atmosphere, in pounds per square inch ....
38. Mean effective pressure in pounds per square inch ....
39. Equivalent M.E.P. in pounds per square inch:
    (a) Referred to first cylinder................................
    (b) Referred to second cylinder...............................
    (c) Referred to third cylinder................................
40. Pressures above zero in pounds per square inch:
    (a) Near cut-off..
    (b) Near release..
    (c) Near beginning of compression.............................
    Percentage of stroke at points where pressures are measured:
    (a) Near cut-off..
    (b) Near release..
    (c) Near beginning of compression.............................
41. Steam accounted for by indicator in pounds per I.H.P. per hour:
    (a) Near cut-off..
    (b) Near release..
42. Ratio of expansion (page 252):
    (a) Commercial..
    (b) Ideal...

SPEED

43. Revolutions per minute. ....

POWER

44. Indicated horse power developed by main engine cylinders:
    First cylinder..
    Second cylinder...
    Third cylinder..
    Total...
45. Brake horse power developed by engine.........................

STANDARD EFFICIENCY AND OTHER RESULTS

46 Heat units consumed by engine and auxiliaries per hour:
  (a) Per indicated horse power, B.T.U. . . . .
  (b) Per brake horse power, B.T.U. . . . .
47. Equivalent standard coal in pounds per hour:
  (a) Per indicated horse power, pounds. . . . .
  (b) Per brake horse power, pounds. . . . .
48. Heat units consumed by main engine per hour corresponding to ideal maximum temperature of feed water given in item 26:
  (a) Per indicated horse power, B.T.U. . . . .
  (b) Per brake horse power, B.T.U. . . . .
49. Dry steam consumed per indicated horse power per hour:
  (a) Main cylinders, including jackets, pounds. . . . .
  (b) Auxiliary cylinders, pounds. . . . .
  (c) Engine and auxiliaries, pounds. . . . .
50. Dry steam consumed per brake horse power per hour:
  (a) Main cylinders, including jackets, pounds. . . . .
  (b) Auxiliary cylinders, pounds. . . . .
  (c) Engines and auxiliaries, pounds. . . . .
51. Percentage of steam used by main engine cylinders accounted for by indicator diagrams, near cut-off of high pressure cylinder, %.

ADDITIONAL DATA

Add any additional data bearing on the particular objects of the test or relating to the special class of service for which the engine is used. Also give copies of indicator diagrams nearest the mean, and the corresponding scales.

Heat Balance. The importance of checking tests of engines cannot be too strongly stated. An important aid for determining the correctness of such tests is obtained by calculating a heat balance, which is making a balance sheet showing the heat received and rejected by the engine. Only such tests should be considered satisfactory which show a reasonable agreement in the heat balance. In ordinary commercial testing by experienced engineers a heat balance is not often calculated, but for accurate laboratory work it should always be made out. It shows at a glance how much heat energy is received by the engine and what disposition is made of it.

In a heat balance the **heat supplied to the engine** will be accounted for in the following items: (1) Heat equivalent of useful work as calculated from brake horse power. (2) Heat equivalent of engine friction. (3) Heat discharged in the condensed steam (into hot well). (4) Heat absorbed by cooling water. (5) Heat radiated and other losses (by difference).

Heat supplied to the engine per minute $\mathbf{Q}_o$ = pounds of steam supplied to engine per minute times the total heat in a pound of steam $(\mathbf{q}_1 + \mathbf{x}_1 \mathbf{r}_1)$, for wet steam and $\mathbf{H}_1 + \mathbf{c}_p(\mathbf{t}' - \mathbf{t}_1)$ for superheated steam. The terms $\mathbf{q}_1$, $\mathbf{x}_1$, $\mathbf{r}_1$, $\mathbf{H}_1$ and $\mathbf{t}_1$ represent as in other equations in this book respectively the heat of the liquid, the quality, the heat of vaporization, the total heat, and the temperature, **all corresponding to the pressure** of the steam supplied. The other term $\mathbf{t}'$ is the temperature of the superheated steam as observed by a thermometer in the steam pipe.

Heat equivalent of the useful work done $(\mathbf{Q}_u)$ per minute is the product of the brake horse power (**B.H.P.**) and the constant 33,000 divided by the mechanical equivalent of a B.T.U.; that is,

$$\mathbf{Q}_u = \frac{\mathbf{B.H.P.} \times 33{,}000}{778} \quad . \quad . \quad . \quad . \quad . \quad (67)$$

Similarly, the heat equivalent of the engine friction $(\mathbf{Q}_f)$ per minute is expressed by

$$\mathbf{Q}_f = \frac{\mathbf{F.H.P.} \times 33{,}000}{778} \quad . \quad . \quad . \quad . \quad . \quad (68)$$

where **F.H.P.** is the friction horse power or the difference between the indicated [1] and the brake horse power.

Heat discharged in the exhaust to the hot-well may be represented by

$\mathbf{Q}_e$ = pounds of steam per minute times the heat of the liquid at the temperature of the exhaust. Or for practical purposes this is the same as taking the product of the pounds of steam per minute and the temperature of the exhaust (condensed steam) in degrees Fahrenheit less 32.

The heat absorbed in the cooling water per minute is then

$\mathbf{Q}_c$ = pounds of cooling water per minute times the difference between the heat of the liquid at the outlet and inlet. For the small difference in temperature occurring in the cooling water of engine tests we may take in place of the difference of the heats of the liquid the difference of the corresponding temperatures, or approximately,

[1] Often it is preferred to express the indicated work in the heat balance instead of the two items useful work and engine friction. Then the heat equivalent of the indicated work $\mathbf{Q}_i = \dfrac{\mathbf{I.H.P.} \times 33{,}000}{778}$.

Q_c = pounds cooling water per minute times the difference between outlet and inlet temperatures.

Heat radiated and other Q_r losses is found, by taking the difference between the heat supplied Q_o and the sum of the several items Q_u, Q_f, Q_e and Q_c.

There is **no "standard" method** for calculating a heat balance. Some engineers prefer to take for the **base** of comparison; that is, **the heat supplied,** the total heat of the steam less q_2, the heat of the liquid in the condensed steam (at the temperature of the exhaust). With this value for the heat supplied the item Q_e drops out and the **net heat supplied,**

$$Q_n = q_1 + x_1 r_1 - q_2 = Q_u + Q_f + Q_e + Q_r.$$

HEAT BALANCE

	B.T.U. per min.		B.T.U. per min.	Per cent.
Heat supplied to the engine, Q_o	......	Heat equivalent of Useful work Q_u	......	......
	......	Heat equivalent Engine friction Q_f	......	......
	......	Heat discharged in exhaust Q_e	......	......
	......	Heat absorbed by Cooling Water Q_c	......	......
	......	Heat Radiated and Other losses Q_r	......	......
Total,		Total,		100

Obviously the totals on the left- and right-hand sides of this balance sheet must be the same.

Thermal Efficiency. The ratio of the heat converted into work to the heat supplied to the engine is called the thermal efficiency. The first of these quantities is calculated from the indicated horse power and the latter from the weight and total heat per pound of the steam supplied. The only uncertainty arises as to the proper **base** from which to calculate. The total heat of steam given in steam tables is calculated from 32 degrees Fahrenheit, but it is obvious if this **base** is adopted the engine may be charged with more than its share of heat. If, for example, the exhaust steam from the engine passes through a feed-water heater and that the engine returns the condensed steam to the boiler as feed-water at say 150 degrees Fahrenheit,

then there will be 150 − 32, or 118 B.T.U. in every pound of steam passing continually from the engine to the boiler and from the boiler back to the engine without doing any work. More accurately, then, the thermal efficiency of an engine ($\mathbf{E}_t$) should be stated as

$$\mathbf{E}_t = \frac{\mathbf{Q}_u + \mathbf{Q}_f}{\mathbf{Q}_n}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (69)$$

where $\mathbf{Q}_u$ and $\mathbf{Q}_f$ as before are the heat equivalents respectively of the useful work and of the engine friction, while $\mathbf{Q}_n$ should be defined as the **net heat supplied** to the engine. From this discussion it follows that the more efficient the feed-water heater is the higher the efficiency of the engine will be, and if an efficient heater is not used there is no reason for charging a loss to the engine. Obviously the limit to the amount of heat returnable to the boiler by means of a heater is the amount of heat in the exhaust steam, and this limit can be nearly approached under actual practical conditions. It is, therefore, very reasonable that the "datum" for the calculation of the heat supplied to the engine should be the heat of the liquid corresponding to the temperature of the steam in the exhaust pipe from the engine. All this applies, of course, equally well to engines whether operating condensing or non-condensing. In other words, the net heat supplied to the engine is the total heat of the steam entering the engine, less the heat of the liquid at the temperature of the engine exhaust.

The temperature of the exhaust must be taken by a thermometer in a suitable thermometer cup placed in the **exhaust pipe** close to the engine; and, similarly, the temperature and quality of the steam supplied to the engine must be determined by thermometers and a steam calorimeter placed close to but on the boiler side of the engine throttle valve.

Engine Performance Expressed in British Thermal Units. A method which is rapidly gaining in favor among practical engineers is to express the performance of a steam engine or turbine by the number B.T.U., supplied, per indicated horse power per minute, the heat units supplied per minute being determined, as explained in the preceding paragraphs; that is, the total heat in the steam entering at the throttle less the heat of the liquid at the temperature of the exhaust. (See page 287.)

Engine Performance Compared with the Rankine Cycle. In order to know how much the efficiency of an engine can be improved it is most desirable to compare the actual thermal efficiency as defined on page 259, with the highest possible efficiency. For steam engines the standard cycle for comparison is now generally taken as the Rankine or Clausius cycle,[1] in which the operation of the engine is assumed to be perfect; that is, without clearance in the cylinder, initial condensation, leakage or radiation. The indicator diagram for the Rankine cycle is represented by **Fig. 190.** The steam is assumed to be supplied to the engine cylinder at constant pressure until the point of cut-off, after which it is expanded **adiabatically** down to the back pressure at which the engine is operated on the return stroke

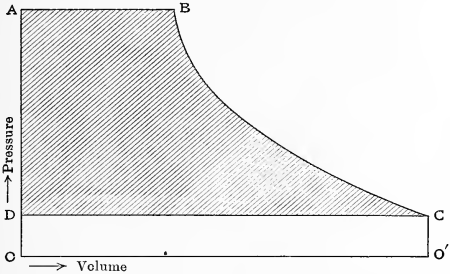


FIG. 190.—Indicator Diagram for the Ideal Rankine or Clausius Cycle.

when the exhaust steam is swept out of the cylinder and returned as feed-water to the boiler at the temperature of the exhaust. The same Rankine cycle represented in **Fig. 190** when shown by a so-called **entropy-temperature diagram** can be made simpler both for analysis and calculations. This other kind of diagram, the details of which are somewhat more difficult to understand, is universally used by steam turbine engineers and has for the problem in hand particular advantages. In this diagram, which will be now described, any surface represents accurately

[1] Years ago it was not unusual to make this comparison with the efficiency of the Carnot cycle as a basis. This efficiency of a heat engine, it will be remembered, is expressed by the ratio of $T_1 - T_2$ to T_1 where T_1 is the absolute initial temperature and T_2 is the absolute final temperature.

to given scales, a quantity of heat. Absolute temperatures (T) are the ordinates, and entropies[1] (ϕ) are the abscissas.

Fig. 191 shows a simple heat diagram laid out with absolute temperature and entropy for the co-ordinates. Steam at a certain condition of temperature and entropy is represented here by the point **A.** Then if some heat is added, increasing both temperature and entropy, the final condition is represented by the point **B,** and the **area ABCD represents the heat added in passing from the condition at A to the condition at B.** Such a diagram is usually called an **entropy-temperature diagram,** although the name **heat diagram** would probably be more appropriate, since **every area represents a definite amount of heat.**

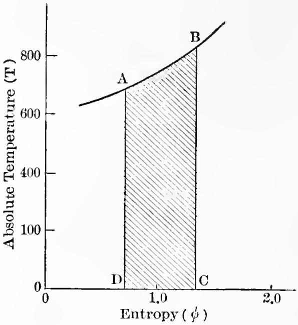


FIG. 191.—A Simple Entropy-temperature Diagram.

Another entropy-temperature diagram is shown in **Fig. 192, representing by the various shaded areas the heat added to water at 32 degrees Fahrenheit to completely vaporize it at the pressure P_1.** The unshaded area under the irregular curve **AB** represents the heat in a pound of water at the freezing point (32 degrees Fahrenheit or 492 degrees in absolute temperature). The area **OBCD** is the heat added to the water to bring it to the temperature of vaporization, or in other words, this last area represents the heat of the liquid (**q**) given in the steam tables for the pressure P_1. Further heating after

[1] Entropy, which Perry calls the "ghostly quantity," has no real physical significance, so that complete definition is not possible. If dQ is a small amount of heat added to a body and T is the absolute temperature at which the heat is added; then the change in entropy of that body is dQ/T, or $d\phi = dQ/T$. Entropy of saturated steam above the entropy of water at the freezing point is easily calculated. For saturated steam at any pressure, then $\phi = xr/T + \theta$, where x is the quality of the steam, r is the heat of vaporization, T is the absolute temperature, and θ is the entropy of the liquid (water).

The symbols used here are those given in Peabody's **Steam and Entropy Tables** (1909).

vaporization begins is **at the constant temperature** T_1 corresponding to the pressure P_1, and is represented by an increasing area under line **CE.** When "steaming" is complete, the latent heat, or the heat of vaporization (r), is the area **DCEF.** If after all the water is vaporized more heat is added, the steam becomes **superheated,** and the additional heat required would be represented by an area to the right of **EF.**

The use of the entropy-temperature diagram in exhibiting the behavior of steam during expansion will now be discussed and illustrated with a practical example.

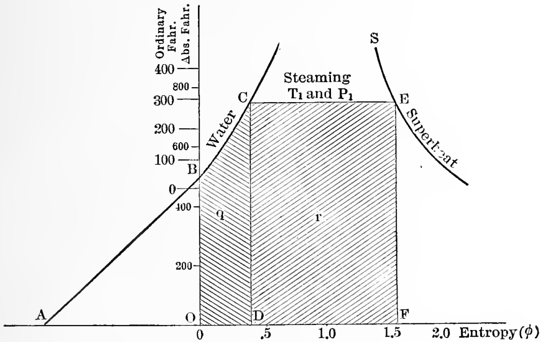


FIG. 192.—Entropy-temperature Diagram Showing Total Heat in a Pound of Dry Saturated Steam.

Fig. 193 illustrates the heat process going on when feed-water is received in the boilers of a power plant at 100 degrees Fahrenheit, is heated and converted into steam at a temperature of 400 degrees Fahrenheit, and then loses heat in doing work. When the feed-water first enters the boiler its temperature must be raised from 100 to 400 degrees Fahrenheit, before any "steaming" begins. The heat added to the liquid is the area **MNCD.** This area represents the difference between the heat of the liquid of steam at 400 degrees Fahrenheit (q_c) and at 100 degrees Fahrenheit (q_n) and is about 306 B.T.U. The horizontal or entropy scale shows that the difference in entropy between water at 100 and 400 degrees Fahrenheit, is about .436.[1]

[1] As actually determined from Peabody's *Steam Tables*, pp. 2 and

Every reader should understand how such a diagram is constructed and especially how the curves are obtained. In this case the **curve NC is constructed** by plotting from the steam tables the values of the entropy of the liquid (usually marked with the symbol θ) for a number of different temperatures between 100 and 400 degrees Fahrenheit.

If now water at 400 degrees Fahrenheit is converted into steam at that temperature, the curve representing the change is necessarily a constant temperature line and therefore a

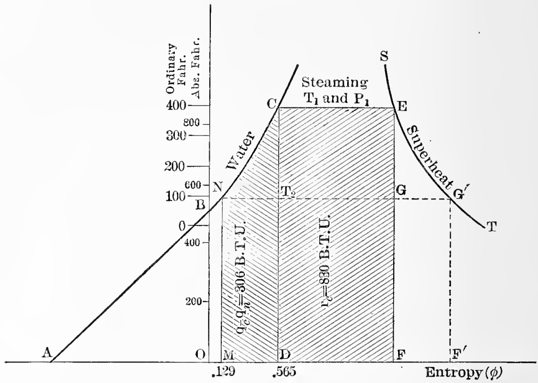


FIG. 193.—Entropy-temperature Diagram Representing Heat Added Above Feed-water Temperature.

horizontal, **CE.** Provided the vaporization has been complete the heat added in the "steaming" process is the latent heat or heat of vaporization of steam (r) at 400 degrees Fahrenheit, which is approximately 830 B.T.U.

The **change in entropy during vaporization** is, then, the heat units added (830) divided by the absolute temperature at which the change occurs (400 + 460 = 860 degrees Fahrenheit absolute) or

$$\frac{r}{T} = \frac{830}{860} = .965.$$

10, the difference in entropy is .565 − .129, or .436. Practically it is impossible to construct the scales in this small figure very accurately.

The **total entropy of steam** completely vaporized at 400 degrees Fahrenheit, is, therefore, the sum of the entropy of the liquid (water) .565 and the entropy of the steam .965, or 1.530.[1] To represent then by **CE** this condition of the steam, the point **E** is plotted where entropy measured on the horizontal scale is 1.530, as shown in the figure.[2] The area **MNCEF** represents then the total heat added to a pound of feed-water at 100 degrees Fahrenheit, to produce steam at 400 degrees Fahrenheit, and the area **OBCEF** represents, similarly, the total heat (**H** in the steam tables) in a pound of steam at 400 degrees Fahrenheit above that in water at 32 degrees Fahrenheit.

Adiabatic Expansion and Available Energy. The practical example illustrated by **Fig. 194** will also be used to explain how the entropy-temperature diagram can be used to show how much work can be obtained by a theoretically perfect engine from the **adiabatic expansion** of a pound of steam. When steam expands adiabatically—without a gain or loss of heat—its temperature falls. Remembering that areas in the entropy-temperature diagram represent quantities of heat and that **in this expansion there is no exchange of heat,** it is obvious that the **area under a curve of adiabatic expansion must be zero;** and this condition can be satisfied only by a vertical line which is a line of constant entropy.[3] For the case in **Fig. 194,** the expansion curve will lie, therefore, **along** the line **EF,** and if the temperature falls to 100 degrees Fahrenheit, the expansion will be from **E** to **G,** and during this change some of the steam has been condensed. If now heat is removed from this mixture

[1] Entropy, like the total heat (H), and the heat of the liquid (q), is measured above the condition of freezing water (32 degrees Fahrenheit).

[2] The point E is shown located on another curve ST, which is determined by plotting a series of points calculated the same as E, but for different pressures. If more heat had been added than was required for vaporization, the area $DCEF$ would have been larger and E would have fallen to the right of ST, indicating by its position that the steam had been superheated. The curve ST is therefore a "boundary line" between the saturated and superheated conditions. This curve can also be plotted from the values obtained from a table of the entropy of dry saturated steam.

[3] Since in an adiabatic expansion there is no change of entropy, lines of constant entropy, in practice, are often called "adiabatics." It is very rare in steam turbine work that an expansion departs far from the adiabatic.

of steam and water till all the steam is reduced to the liquid state, **but without further lowering of the temperature,** the horizontal line **GN** [1] will present the change in its condition. The quantity of heat rejected in this last process—technically known as **condensing the steam**—is represented by the area **MNGF,** and the **heat converted into work** is, therefore, the area **NCEG**; and this is called the **available energy.** By means of

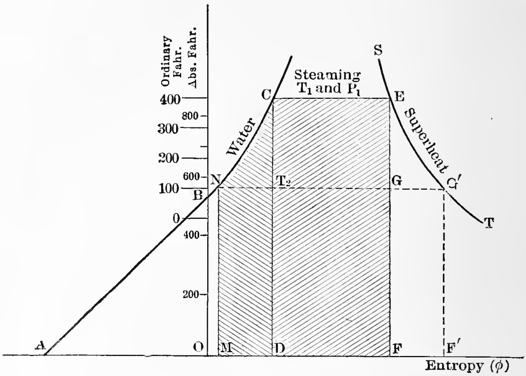


FIG. 194.—Entropy-temperature Diagram Illustrating the "Available Energy" in Steam.

diagrams like those in the preceding figures, it will now be shown how the **available energy of dry saturated steam** for any given

[1] That the steam might be dry and saturated, the expansion would have had to follow the curve ET and G would have appeared at G'.

The heat of the liquid, q, of a pound of steam at 100 degrees Fahrenheit is represented by $OBNM$, and the heat of vaporization (r) is $MNG'F'$, so that the total heat ($q+r$ or H) is $OBNG'F'$. The total heat of wet steam is expressed by $q+xr$, where x is the quality or relative dryness. In the case of this adiabatic expansion, then, q is as before $OBNM$ and xr is $MNGF$. It is obvious also that the lines NG and NG' have the same relation to each other as the areas under them, so that

$$\frac{\text{line } NG}{\text{line } NG'}=\frac{\text{area } MNGF}{\text{area } MNG'F'}=\frac{xr}{r}, \text{ or } \frac{NG}{NG'}=x, \quad . \quad . \quad . \quad (70)$$

showing that the quality of the steam at any point, G, on a constant temperature line (which for saturated steam is also a constant pressure line) is determined as in this case by the ratio of NG to NG'.

conditions can be readily calculated from the data given in steam tables.

Fig. 195 is an entropy-temperature diagram representing dry saturated steam which is expanded adiabatically from an initial temperature T_1 corresponding to a pressure P_1 to a lower final temperature T_2 corresponding to a pressure P_2. The other initial and final conditions of total heat (H) and entropy (ϕ) are represented by the same subscripts 1 and 2. The available energy or the work that can be done by a perfect engine under these conditions is the area NCEG. It is now desired to obtain a simple equation expressing this available energy E_a in terms of total heat, absolute temperature and entropy. Explanations of the preceding figures should make it clear that

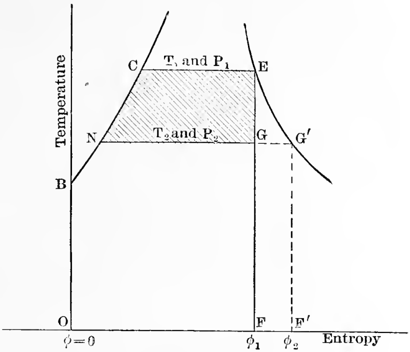


Fig. 195.—Practical Example of Adiabatic Expansion.

$$H_1 = \text{area } OBNCEGF,$$

$$H_2 = \text{area } OBNG'F',$$

$$E_a = \text{areas } (OBNCEGF + FGG'F') - OBNG'F',$$

$$E_a = H_1 - H_2 + FGG'F',$$

therefore

$$E_a = H_1 - H_2 + (\phi_2 - \phi_1)T_2{}^1. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (71)$$

An application of this equation will be made at once to determine the heat energy available from the adiabatic expansion of a pound of dry saturated steam at an initial pressure

[1] It should be observed that this form is for the case where the steam is initially dry and saturated. For the case of superheated steam a slightly different form is required which is given on page 272.

of 165 pounds per square inch absolute to a final pressure of 15 pounds per square inch absolute.

Example. $P_1 = 165$ $T_1 = \ldots$
$P_2 = 15$ $T_2 = 673.0$ from steam tables.[1]
$H_1 = 1193.6$ from steam tables.
$H_2 = 1146.9$ from steam tables.
$\phi_1 = 1.5605$ from steam tables.
$\phi_2 = 1.7499$ from steam tables.

Substituting these values in equation (71), we have

$$E_a = 1193.6 - 1146.9 + (1.7499 - 1.5605)673.0 = \mathbf{174.2} \quad \textbf{B.T.U.}$$

per pound of steam.

The **important condition assumed** as the basis for the deter-

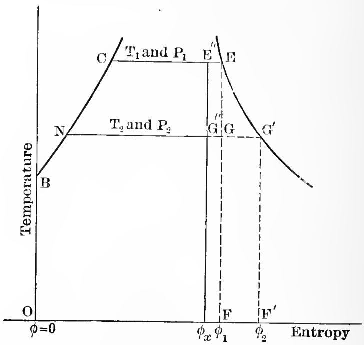


FIG. 196.—Entropy-temperature Diagram of Initially Wet Steam.

mination of equation (71), that the **steam is initially dry and saturated,** must not be overlooked in its application. There are, therefore, two other cases to be considered:

(1) When the steam is initially wet.

(2) When the steam is initially superheated.

Available Energy of Wet Steam. The case of **initially wet steam** is easily treated in the same way as dry and saturated steam.

[1] The values of the properties of steam given in the exercises are taken from Peabody's "Steam and Entropy Tables." (1909.)

Fig. 196 is an example of the case in hand. At the initial pressure $\mathbf{P_1}$, the total heat of a pound of wet steam $(\mathbf{q_1}+\mathbf{x_1 r_1})$ is represented in this diagram by the area **OBNCE″**$\boldsymbol{\phi}_x$. The initial quality of the steam $(\mathbf{x_1})$ is represented by the ratio of the lines $\frac{\mathbf{CE''}}{\mathbf{CE}}$. The available energy from adiabatic expansion from the initial temperature $\mathbf{T_1}$ (corresponding to the pressure $\mathbf{P_1}$) to the final temperature $\mathbf{T_2}$ (corresponding to the pressure $\mathbf{P_2}$) is the area **NCE″G″**. If we call this **available energy** $\mathbf{E}_{aw}$, we have

$$\mathbf{E}_{aw} = \text{area } \mathbf{OBNCEGF} + \mathbf{FGG'F'} - \mathbf{OBNG'F'} - \mathbf{G''E''EG},$$

$$\mathrm{E}_{aw} = \mathrm{H_1} - \mathrm{H_2} + (\phi_2 - \phi_1)\mathrm{T_2} - (\phi_1 - \phi_x)(\mathrm{T_1} - \mathrm{T_2}),^1$$

$$\mathbf{E}_{aw} = \mathbf{H_1} - \mathbf{H_2} + (\boldsymbol{\phi}_2 - \boldsymbol{\phi}_1)\mathbf{T_2} - \frac{\mathbf{r_1}}{\mathbf{T_1}}(1 - \mathbf{x_1})(\mathbf{T_1} - \mathbf{T_2}). \quad . \quad . \quad (72)$$

Example. Calculations of the heat energy from adiabatic expansion for the same conditions given in the preceding example on page 268, except that the steam is initially 5 per cent wet, are given below.

$\mathbf{P_1} = 165$ lbs. abs. $\quad \mathbf{T_1} = 825.9°$ F.

$\mathbf{P_2} = 15$ lbs. abs. $\quad \mathbf{T_2} = 673.0°$ F.

$\mathbf{H_1} = 1193.6$ B.T.U.

$\mathbf{H_2} = 1146.9$ B.T.U.

$\boldsymbol{\phi}_1 = 1.5605.$

$\boldsymbol{\phi}_2 = 1.7499.$

$\mathbf{r_1} = 855.9$ B.T.U.

$\mathbf{x_1} = 1.00 - .05 = .95.$

$$\mathrm{E}_{aw} = 1193.6 - 1146.9 + (1.7499 - 1.5605)673.0 - \frac{855.9}{825.9} \times .05(825.9 - 673.0),$$

$\mathbf{E}_{aw} = \mathbf{166.3}$ **B.T.U.**

[1] In general terms, $\quad \phi = \frac{xr}{T} + \theta.$ Here

$$\phi_1 = \frac{r_1}{T_1} + \theta_1 \quad \text{because } x = 1.$$

$$\phi_x = \frac{x_1 r_1}{T_1} + \theta_1.$$

$$\phi_1 - \phi_x = \frac{r_1}{T_1}(1 - x_1).$$

Specific Heat of Superheated Steam. In modern practice, superheated steam often enters our calculations, and a troublesome modification of the entropy diagram results. The difficulty arises because the specific heat of superheated steam is not very accurately known. The diagrams following are calculated for the specific heat determinations by Knoblauch and Jakob.[1] The specific heat of steam varies with the temperature and pressure as shown in **Figs. 197** and **198**, giving

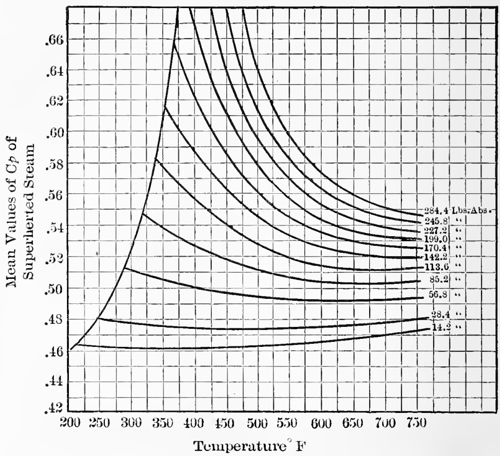


FIG. 197.—Mean Values of Specific Heat (C_p) of Superheated Steam Integrated from Knoblauch and Jacob's Data.

values of the **mean** and the **true** specific heat at constant pressure (C_p).

True specific heat represents the ratio of the amount of heat to be added to a given weight of steam at some particular condition of temperature and pressure to raise the temperature one degree to that required to raise the temperature of water at maximum density one degree. The **mean specific heat** is almost invariably used in steam engine and turbine calculations.

[1] *Zeit. Verein deutscher Ingenieure*, Jan. 5, 1907. Values of mean specific heat are taken from *Mechanical Engineer*, July, 1907, and Professor A. M. Greene's paper in *Proc. American Society of Mechanical Engineers*, May, 1907.

Entropy Diagram of Superheated Steam. The graphic representation of the heat added during the superheating of steam is easily shown with entropy-temperature diagrams. **Fig. 199** shows a diagram similar to the one representing dry saturated steam with the added area **EHJF** to show the super-

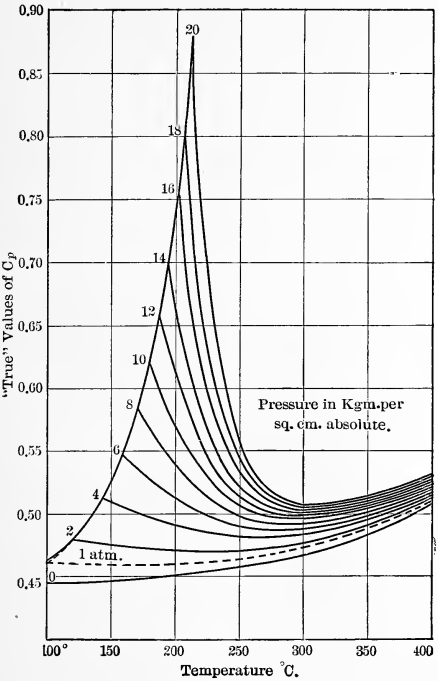


FIG. 198.—Values of the " True " Specific Heat of Superheated Steam.

heating from the temperature, T_1 corresponding to the pressure P_1 to the temperature of the superheated steam, T_s. The total heat in a pound of steam above the freezing point is now represented by the area **OBCEHJFO**. For adiabatic expansion of superheated steam at the temperature T_s and pressure P_1 to a pressure P_2 the **available energy is the area CEHKL.**

Too much calculation is involved in the construction of entropy diagrams to make a new diagram for every particular case from the properties usually found in steam tables; but the construction of such diagrams should be understood. From the explanations that have preceded, the construction of all the lines except EH should be obvious. This line is obtained by calculating the entropy of superheated steam for various values of temperature from the following well-known relations:

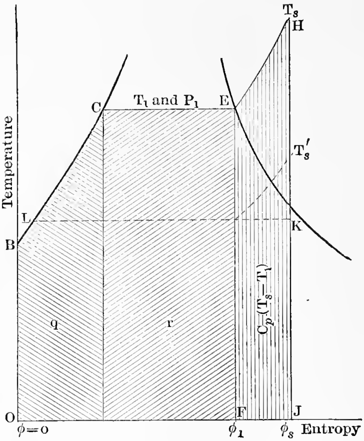


FIG. 199.—Entropy-temperature Diagram for Superheated Steam.

$$\phi_s-\phi_1=\int\frac{dQ}{T}=\int_{T_1}^{T_s}\frac{C_p dT}{T},$$

or

$$\phi_s-\phi_1=C_{pm}\left[\log_e\frac{T_s}{T_1}\right]=2.3028\,C_{pm}\left(\log_{10}(T_s-\log_{10}T_1)\right), \quad (73)$$

and

$$\mathbf{E}_{as}=\mathbf{H}_1-\mathbf{H}_2+\mathbf{C}_{pm}(\mathbf{T}_s-\mathbf{T}_1)-\mathbf{C}_{pm}(\mathbf{T}_s'-\mathbf{T}_2) \quad . \quad . \quad (74)$$

where $\mathbf{E}_{as}$ is the available energy of the Rankine cycle for superheated steam and $\mathbf{T}_s'$ is calculated from the equation above where ϕ_s and ϕ_2 $(=\phi_1)$ are now both known quantities. $\mathbf{C}_{pm}$ is the **mean value** taken from the curves in **Fig. 197** for the temperature $\mathbf{T}_s$, and $\mathbf{C}'_{pm}$ for $\mathbf{T}_s'$.

Approximate Steam Consumption Calculated from an Indicator Diagram. It is often very convenient to be able to calculate the approximate steam consumption of a steam engine from the data obtainable from an indicator card, the size of the piston,

the stroke, and the speed. Using a double-acting engine, the following symbols [1] may be used:

p =mean effective pressure, pounds per square inch from indicator diagram.
l =length of the stroke of the engine in **feet.**
a =area of the piston in square inches.
b =percentage of clearance to the length of the stroke.
c =percentage of stroke at any point in the expansion line.[2]
n =number of revolutions per minute; and **120 n** =number of strokes per hour.
w =weight of a cubic foot of steam having a pressure as shown by the indicator diagram corresponding to that at the point in the expansion line selected for **c,** pounds.
w′ =weight of a cubic foot of steam corresponding to the pressure at the **end** of compression, pounds.

Then the number of cubic feet per stroke $=\frac{la(b+c)}{144(100)}$ in the clearance and piston displacement volumes (at **c**).

$$\text{Weight of steam per stroke, pounds} = \frac{law(b+c)}{144(100)}. \tag{75}$$

$$\text{Volume of the clearance, cubic feet} = \frac{la(b)}{144(100)}.$$

Weight of steam in clearance, pounds remaining in the cylinder $=\frac{law'(b)}{144(100)}$.

Approximate net weight of steam used per stroke

$$=\frac{law(b+c)}{144(100)}-\frac{law'(c)}{144(100)}=\frac{la}{14,400}\Big[(b+c)w-cw'\Big]. \tag{76}$$

Approximate weight of steam from diagram per hour

$$=\frac{120\ nla}{14,400}\Big[(b+c)w-cw'\Big]. \quad . \quad . \quad . \tag{77}$$

[1] Compare with *Power*, September, 1893.

[2] In other words this is the percentage of the entire stroke which has been swept through by the piston corresponding to the point in the expansion curve selected for measurements. It is preferable, however, to take this point not very far from the point of cut-off, since the assumption must be made that the product of pressure times volume in the expansion curve is constant, which is, of course, not accurate, and the error becomes greater as the expansion increases.

Indicated horse power for a double acting engine

$$=\frac{2plan}{33,000}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (78)$$

Steam consumption per indicated horse power is (77) divided by (78) or

$$=\frac{137.5}{p}\Big[(b+c)w-cw'\Big]. \quad . \quad . \quad . \quad . \quad . \quad . \quad (79)$$

The difference between the theoretical steam consumption calculated by the formula and the actual consumption as determined by tests represents steam "not accounted for by the indicator," due to cylinder condensation, leakage through ports, radiation, etc. If the steam supplied to the engine is very wet, corrections for this moisture should be made in the value of w.

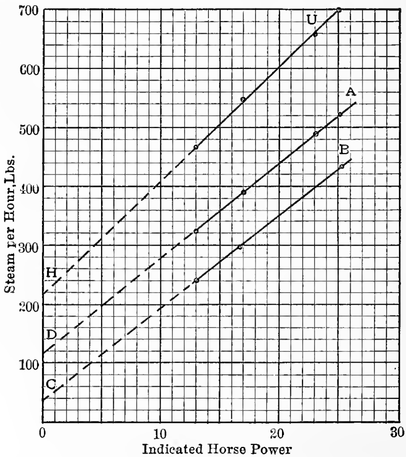


FIG. 200.—"Willans" Lines for an Engine with a Throttling Governor.

Willans Law. One of the most serviceable **checks** that can be applied to engine tests is plotting the **Willans line** of total steam consumption per hour. Curve sheets illustrating this as plotted from actual tests by Barraclough & Marks [1] are shown in **Fig. 200.** It will be observed that the points representing the weight of steam used per hour when plotted for the horse power corresponding are on a **straight line.** In other words Willans law is usually stated thus: "With a fixed cut-off

* *Proceedings Institution of Civil Engineers*, vol. 120, page 323.

and a **throttling governor** the total steam used by the engine per hour at different loads can be represented by a straight line upon a mean effective pressure base or upon a horse-power base." It will be shown also in the following paragraphs how, theoretically, this relation holds for an engine operating at a fixed cut-off and with a throttling governor and that the steam consumption per hour is proportional to the mean effective pressure and also to the horse power developed.

If we assume that the expansion curve is hyperbolic, which is usually near the truth, then the mean "forward" pressure given by an indicator diagram is [1]

$$p_1\left(\frac{1+\log_e r}{r}\right), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (80)$$

where p_1 is the initial pressure of the steam and **r** is the ratio of expansion. With a **throttling** governor **r** is of course constant. The terms in the parentheses can then be represented by a constant **c** and the mean forward pressure p_f then can be written as p_1c. Volume of steam used per hour for a **double-acting** engine can be expressed in cubic feet as

$$120(nV), \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (81)$$

where **n** is the number of revolutions per minute and **V** is the volume of steam admitted to the cylinder per stroke. Now if we use the symbol v_1 for the specific volume, that is, the volume in cubic feet of a pound of steam at the pressure p_1, and assuming for a small range of pressures that p_1v_1 = a constant **k**, then we can write, if **W** is the weight of steam used per hour in pounds,

$$W=\frac{120(nV)}{v_1}=\frac{120(nVp_1)}{k}. \quad . \quad . \quad . \quad . \quad . \qquad (82)$$

Now with a throttling governor and constant cut-off all these quantities are constant except p_1, and writing a constant **z** for the term $\frac{120(nV)}{k}$ we have $W=zp_1$, but it was shown above that the mean **forward** pressure $p_f=cp_1$, so that

$$W=z\frac{p_f}{c}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (83)$$

[1] Compare with Perry's "Steam Engine," page 286.

So that the curve representing this equation is a **straight line** and passes through the origin of co-ordinates. If, however, we use the mean **effective** pressure instead of the mean forward pressure, then

$$\text{M.E.P.} = p_f - p_b, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (84)$$

where p_b is the mean "back" or exhaust pressure. In these last terms then

$$W = \frac{z}{c}(\text{M.E.P.} + p_b). \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (85)$$

This last equation may be stated as $W =$ a constant $\times$M.E.P.+ another constant which, when plotted to a scale of mean effective pressure for abscissa and weight of steam used per hour for ordinates, is also a **straight line,** intersecting the axis of ordinates above the origin at a distance corresponding to the steam consumption per hour at no load.

Since the **indicated horse power** (**I.H.P.**) is proportional to mean effective pressure, a straight line will result when the steam consumption and indicated horse power are plotted; and the same holds true also when steam consumption is plotted with **brake horse power** (**B.H.P.**) instead of indicated horse power.

Curves Showing Results of Tests Graphically. One of the best checks of an engine test is to plot the principal observations graphically as the test proceeds. This is particularly important as regards the **total weight of steam used** per hour. For a series of tests each made with a different load the points plotted with horse power (either indicated or brake) as abscissas and total steam per hour as ordinates should lie along a **straight line,** known as the **Willans line** (see page 274). This statement applies accurately only for engines operating with a throttling governor at, of course, constant speed; but is generally applicable to steam turbine tests, irrespective of the type of governor.

Steam Engine Lubricators. The proper oiling of an engine is most important. The operation of nearly all types of lubricating devices is easily understood, particularly when operated by the **gravity of the oil** or by a pump. Another type of lubricator for cylinder lubrication which is operated by the weight

of a column of **condensed steam** is shown in **Fig. 201.** The pipe **C,** in which the condensed steam accumulates, must be made at least 2 feet long to give a sufficient head or pressure to feed the oil. The oil reservoir is in the cylindrical vessel below the condenser pipe. This is filled with oil through an opening in the top as the water which has accumulated in the apparatus is drained through the cock **D.** Water from the condenser

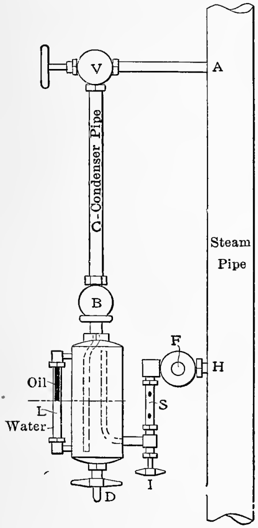


FIG. 201.—Engine Cylinder Lubricator Operated by Pressure of Column of Condensed Steam.

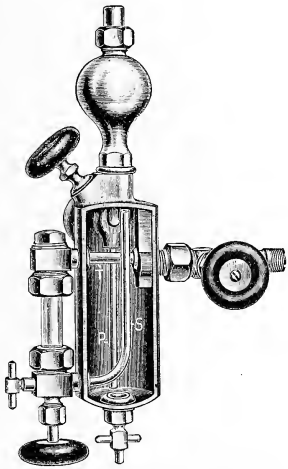


FIG. 202.—"Detroit" Cylinder Lubricator.

pipe **C** is carried down to the bottom of the oil reservoir by the pipe shown in dotted lines on the left-hand side. Oil, being lighter than the water, remains in the upper part of the reservoir and is forced down through the pipe on the right-hand side and through the needle-valve **I** by which the flow of oil can be regulated so that as small an amount as one drop in two or three minutes passes into the steam pipe at **H** to mix with the steam going to the engine cylinder. Through the

gage glass **S** the number of drops passing through can be observed. Another gage glass **L** on the side of the reservoir shows the relative amounts of oil and water. **When an engine is not operating all oil cups and lubricators should be carefully closed.** The feed of oil from this lubricator is stopped by closing the valves **V** and **F.** A valve is usually provided on the bulb **B,** which should be closed when draining from **D,** so that the water in the condenser pipe **C** will not be lost, and thus prevent the operation of the apparatus till a sufficient amount of condensed steam has accumulated to produce the pressure necessary to force the oil into the steam pipe.

A slight modification of the lubricator described is illustrated in **Fig. 202.** The condensed steam is brought to the bottom of the reservoir through the pipe **P,** which is open at its lower end. On the other hand, the pipe **S** is open at the top and the oil is forced by the pressure due to the head of water in the pipe above (not shown) through the gage glass on the left-hand side, then through the horizontal pipe **T** to which is attached the valve and nipple connected to the steam pipe supplying the engine. In this figure at the top of the reservoir a plug with a nicely finished handle is shown, which is to be removed for filling with oil.

CHAPTER XII

TESTING STEAM TURBINES AND TURBINE GENERATORS

Testing Steam Turbines.[1] In every power plant the means should be available for making tests of the steam equipment to determine the steam consumption. Usually tests are made to determine how nearly the performance of a turbine approaches the conditions for which it was designed. The results obtained from tests of a turbine are to show usually the steam consumption required to develop a unit of power in a unit of time, as, for example, a horse power hour or a kilowatt hour.

In such tests a number of observations must be made regarding the condition of the steam in the passage through the turbine and of the performance of the turbine as a machine. To get a good idea of what these observations mean, it may be profitable to follow the steam as it passes through the turbine. The steam comes from the boilers through the main steam pipe and the valves of the turbine to the nozzles or stationary blades as the case may be. It then passes through the blades and finally escapes through the exhaust pipe to the condenser. It is preferable to have a surface condenser for tests, so that the exhaust steam can be weighed. The weighing is preferably done in large tanks mounted on platform scales.

Methods for Testing. The important observations to be made in steam turbine tests are:

1. Pressure of the steam supplied to the turbine.

[1] Tests of the turbines alone in a modern station may be only a rough indication of the over-all economy of the plant. Recently steam turbines were installed in a large power plant where they replaced steam engines of an excellent make. Tests of the turbines and of the engines made without considering the losses in the rest of the plant showed very little gain in efficiency by this change, although it was found that the fuel consumption was reduced twenty per cent.

Parts of this chapter and the chapter following are taken from the author's work on "The Steam Turbine."

2. Speed of rotation of the turbine shaft, usually taken in revolutions per minute.

3. Measurement of power with a Prony or a water brake, if the power at the turbine shaft is desired; or with electrical instruments (ammeters, voltmeters, and wattmeters), if the power is measured by the output of an electric generator.

4. Weight, or measurement by volume, of the condensed steam discharged from the condenser. Unless a surface condenser is used it is very difficult to obtain the amount of steam used by the turbine. All leakages from pipes, pumps, and valves, which are a part of the steam which has gone through the turbine, must be added to the weight of the condensed steam. The accuracy of a test often depends a great deal on how accurately leaks have been provided against, or measured when they occur.

5. Temperature of the steam as it enters the turbine. If the temperature is higher than that due to the pressure of the saturated steam given in steam tables, the steam is superheated: if, however, the temperature is not higher the steam may be wet, and a calorimeter must be attached as near the turbine steam chest as possible.[1]

All gages, electrical instruments, and thermometers should be carefully calibrated before and after each test, so that observations can be corrected for any errors. The zero readings of Prony and water brakes for measuring power should be carefully observed and corrected to eliminate the friction of the apparatus with no load. Unless all these precautions are taken the difficulties in getting reliable tests of turbines are greatly increased. In all cases tests should be continued for several hours with absolutely constant conditions if the tests are to be of value.

The most valuable test of a steam turbine or of a reciprocating steam engine is made when varying only the load; that is, with pressures, superheat, and speed constant. When the steam consumption is then plotted against fractions of full load, a

[1] The most satisfactory tests of turbines are made with steam slightly superheated rather than wet. When steam is very wet (more than about 4 per cent moisture for ordinary pressures) the determination of the quality is difficult. There is also a danger that steam showing only a few degrees of superheat by the reading of the thermometer is actually wet. The high temperature is due in such cases to heating from eddies around the thermometer case or in steam pockets near it.

water-rate curve is obtained. For such a curve a series of tests are needed, each for some fraction of full load; and in each separate test the power as well as all the other conditions must be held constant.

Another important test of the performance of a steam turbine is made by varying both the speed and the power and keeping the other conditions constant. The observations of speed and power from such a test give a power parabola as illustrated in **Fig. 203.** This curve shows at what speed the turbine gives the greatest output.

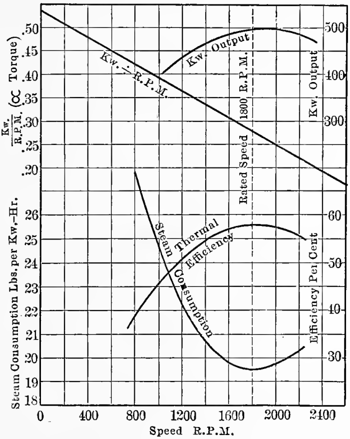


FIG. 203.—Results of Tests of a Turbine at Various Speeds.

Tests may also be made with varying initial steam pressure, but keeping other conditions including exhaust pressure and load constant.

Calculations of the steam consumption and efficiency of turbines made by allowing for the different losses as calculated separately and then added together as is often done to determine the losses in electrical apparatus are of very little value except when made by experienced designers.[1]

Commercial Testing. The methods used by the New York Edison Company in commercial tests of steam turbine-generator units may well be explained briefly.

During a test the load on the turbine unit is maintained as constant as possible by "remote control" of the turbine gov-

[1] This method of calculation of steam consumption is explained in detail in "The Steam Turbine," by the author, pages 86–93. Steam consumption of a turbine can be predicted by calculations much more accurately than for a steam engine.

ernor by the switchboard operator. The maximum variation in load is to be held within 4 per cent above and below the mean. For some time previous to the test the turbine is run a little below the load required for the test, but at least ten minutes before the starting signal is given the test load must be on the machine.

Three-phase electrical load is measured by the two-wattmeter method,[1] using Weston indicating wattmeters of the standard laboratory type. These instruments are calibrated at a well-known testing laboratory immediately before and after the test. Power factor is maintained substantially at unity and all electrical readings are taken at one-minute intervals.

When the turbine is supplied with a surface condenser, the steam consumption, or water rate, is determined by weighing in a large tank supported on platform scales the condensed steam delivered from the condenser hot well. Above the tank on the scales a reservoir is provided which is large enough to hold the condensation accumulating between the weighings, which were made at intervals of five minutes. By using a loop connection for the gland water supply (of Westinghouse turbines) or the water from the step bearing (of Curtis turbines, using water for this bearing) the necessity for connecting the weighings for these amounts is avoided.

Because the circulating water at the stations of this company is usually quite salty, any condenser leakage is detected by testing the condensed steam by the silver-nitrate test with a suitable color indicator. This color method is said to be a decided advantage over the usual method of weighing the leakage accumulating during a definite period when the condenser is idle and tested only with full vacuum. By taking samples of circulating water and condensed steam at the same time, it is possible to detect any change in the rate of condenser leakage.

The water level in the hot well is maintained at practically a constant point by means of a float valve in the well, automatically controlling the speed and, therefore, the amount of the delivery of the hot-well pump. This device avoids the

[1] Cf. Kent's "Mechanical Engineer's Pocket-Book," 8th ed., page 1396, or Foster's "Electrical Engineer's Pocket-Book," 4th ed., pages 51 and 325.

necessity for the difficult correction to be made in a test when the levels in the hot well are not the same at the beginning and end of a test. Temperatures and pressures of the admission steam are determined by mercury thermometers and pressure gages located near the main throttle valve of the turbine; the amount of superheat is determined by subtracting from the actual steam temperature after making thermometer connections the temperature of saturated steam corresponding to the pressure at the point where the temperature is measured. All gages and thermometers are calibrated before and after the test.

Vacuum is measured directly at the turbine exhaust by means of a mercury column with a barometer alongside for reducing the vacuum to standard barometer conditions (30 inches). By this latter arrangement the necessity for **temperature connections** between the two mercury columns not at the same place is avoided.

It has sometimes happened that split condenser tubes have caused a leakage of steam which was extremely difficult to measure. Cases are reported where the split opened up only when the condenser was heated with a large volume of steam. On this account it is preferable not to use a leaky condenser for accurate tests; in other words, the condenser should be thoroughly repaired before tests are made. The effect of split tubes causing an irregular amount of leakage is usually shown in tests by inconsistent results in the weight of condensed steam. In that case the leakage will be greatest with largest flow of steam through the condenser and it will be observed, for an engine operating with a throttling governor or with a steam turbine, that when the "Willans line" page 274, is drawn to **check** the tests that it will be curved instead of straight. It should be noted, however, that a curved "Willans line" does not necessarily indicate this phenomenon in condenser leakage, as the irregularity may be due to faulty design of the engine[1] or turbine. Tests of condensers for leakage should be run long enough so that the quantity of water coming through can be determined with accuracy.[2] Usually a leakage test run for less than an hour or two is of no use at all.

[1] The "Willans" line for a reciprocating engine operating with an automatic cut-off governor is usually a curve slightly concave upward.

[2] To determine accurately the weight of condensed steam the air-

When the method of determining the weight of condensed steam by weighing the boiler feed-water is used the chances of error are very great and every possible precaution to insure accuracy must be observed. In the first place valves no matter how good should not be relied on to prevent the passage of steam through them. For this reason careful engineers insist on disconnecting from the line of steam piping between the boilers and the engine or turbine tested all other piping connected to it, and then blanking off with flanges all the sections disconnected. If flanged pipe fittings have been used in the pipe lines, blanking off sections in the various pipes is very easily accomplished by disconnecting the flanges and inserting a thin iron or copper plate with holes around the edge to fit the bolt-holes in the flanges. The plate is then easily bolted in place. Another important precaution to observe is that the outlets of all drain or drip pipes and of all blow-off valves must be visible. It is equally important that all the piping between the boiler feed-pumps and the boilers is exposed with all branches blanked off or plugged. Boiler leakage should be determined before and after each test with preferably the pipe supplying the turbine blanked off at the throttle valve, although if the throttle valve is reasonably tight the precaution of blanking off this valve is not considered so important as the others mentioned. In tests for boiler leakage the required steam pressure must be maintained on both the piping and the boilers. Measuring feed-water with water meters should not be thought of unless only approximate results are expected, and in that case such an arrangement is only allowable if the temperature of the boiler feed is not over 80 to 90 degrees Fahrenheit for most meters, and a by-pass at the meter is provided, so that at frequent intervals the meter can be calibrated by actual weighing of the flow through it, with the rate of flow and temperature of the water the same as in the tests. Boiler leakage is often as much as from ten to fifteen per cent of the weight of feed-water, and in some reliable tests a still greater leakage has been observed.

pump, piping, and tanks must be free from leaks, and the condenser and pump should be so arranged with respect to each other that the condensed steam will flow in a continuous stream to the pump and into the tanks.

Steam Consumption Determined by a "Heat Balance" Method. There is still another method sometimes used for determining the steam consumption of engines and turbines operated with jet condensers or condensers of a similar type where the cooling water and condensed steam are mixed and discharged together from the condenser. This method is based on the measurement of the amount of heat absorbed by the cooling water from the condensed steam. The weight $\mathbf{w}_c$ and temperature of the cooling water leaving the condenser $\mathbf{t}_2$, the quality of the exhaust steam $\mathbf{x}$ and the temperature of the mixture of condensed steam and cooling water $\mathbf{t}''$ are determined as accurately as possible and from these data the weight of condensed steam $\mathbf{w}_s$ is of course readily calculated by a simple algebraic equation as follows:

$$w_c(t_2-32)+w_s(q+xr)=(w_c+w_s)(t''-32) \quad . \quad . \quad (86)$$

where $\mathbf{q}$ and $\mathbf{r}$ are respectively the heat of the liquid and the heat of vaporization corresponding to the temperature of the exhaust steam. In this equation "heat contents" are measured for each term from 32 degrees F. The method is, however, unreliable, and at best can be depended on for only very approximate results. The reason for this inaccuracy is the difficulty of measuring, especially in large plants, the quantity of cooling water and the true average temperature of a large volume of water flowing in a pipe or channel. It is found usually that the temperature of the water discharged from the condenser will vary from one side of the pipe to the other, and small errors in the determination of this temperature, because the rise in temperature is small, will make large discrepancies in the calculated weight of condensed steam.

Stage pressures should be observed and recorded when tests are made of steam turbines having a series of pressure stages. These data are often extremely useful, both for checking the weight of condensed steam if the turbine is of the nozzle type [1]

[1] Usually the nozzles discharging the steam into the second stage of the turbine are always open, so that the total area is always constant. If, therefore, the areas of the smallest sections of these nozzles are measured and the pressure is observed in the first stage, the weight can be calculated with a considerable degree of accuracy by using the formula for the flow of steam on page 148.

and also for showing any abnormal conditions in the several stages.

Results calculated on a basis of kilowatts output should be **net**; that is, the power required for excitation should be subtracted from the generator output. If, however, the generator is self-exciting the net output can be measured directly at the terminals of the machine.

"Guarantee" tests of steam engines and turbines should be made under conditions as nearly as possible the same as that for which the turbine was designed. Different machines will have different correction factors for varying conditions of pressure, superheat, vacuum, etc., so that water rates corrected for large variations are always likely to be more or less inaccurate. This is particularly true in respect to vacuum corrections. Some turbines will give a very good efficiency with a low vacuum, but at a high vacuum because of an insufficiently large steam space the efficiency will be low.

HEAT UNIT BASIS OF EFFICIENCY

A **thermal efficiency** can be calculated readily by determining what percentage the heat equivalent of the work is of the heat "used by the turbine," assumed to be the difference between the total heat in the steam at the initial conditions and the heat of the liquid in the condensed steam at the temperature of the exhaust.

By this method the full load test of a Westinghouse-Parsons turbine reported by F. P. Sheldon & Co., will be calculated from the data given in an official report.

In order to make the results of such calculations of steam turbine tests comparable with the usual heat unit computations of reciprocating steam engine tests the results are often expressed in terms of indicated or **"internal" horse power.** It was assumed the mechanical efficiency of a reciprocating engine of about the same capacity at this load was about 93.3 per cent.

THERMAL EFFICIENCY OF A 400-KILOWATT STEAM TURBINE

Brake horse power	660
Corresponding indicated or "internal" horse power of a reciprocating engine $= \frac{660}{.933}$	708
Total steam used per hour, pounds	9169
Steam used per "internal" horse power per hour, pounds	12.96
Steam used per "internal" horse power per minute, pounds	0.216
Steam pressure, pounds per square inch, absolute	166.9
Superheat, degrees Fahrenheit	2.9
Vacuum, referred to 30 inches barometer, inches	28.04
Temperature of condensed steam, degrees Fahrenheit (at .96 pound per square inch absolute pressure)	100.6
Total heat contents of one pound of dry saturated steam at the initial pressure, B.T.U.	1193.9
Heat equivalent of superheat in one pound of steam, B.T.U. (C_p from Fig. 197, page 270)	1.9
Total heat contents of one pound of superheated steam, B.T.U.	1195.8
Heat of liquid in condensed steam, B.T.U.	68.6
Heat used in turbine per pound steam, B.T.U.	1127.2
Heat used in turbine per "internal" horse power per minute, B.T.U. (1127.2×0.216)	243.5
Heat equivalent of one horse power per minute, B.T.U. $= \frac{33000}{778}$	42.42
Thermal efficiency, per cent ($42.42 \div 243.5$)	17.4

CALCULATION OF EFFICIENCY (SHAFT AND BUCKET) OF A STEAM TURBINE GENERATOR COMPARED WITH THE RANKINE OR CLAUSIUS CYCLE.

1. "Electrical" kilowatts.
2. R.P.M.
3. Steam per hour (corrected for moisture).
4. Water rate per "electrical" kilowatt, pounds per hour. $(3) \div (1)$
5. I^2R, loss in generator, kilowatts.
6. Rotation loss of generator alone, kilowatts.
7. Rotation loss of wheel and generator, kilowatts.
8. "Shaft" kilowatts. $(1) + (5) + (6)$
9. "Bucket" kilowatts. $(1) + (5) + (7)$
10. Water rate per "shaft" kilowatt, pounds per hour.
11. Water rate per "bucket" kilowatt, pounds per hour.
12. Steam-chest pressure, pounds per square inch, absolute.
13. Exhaust pressure, pounds per square inch absolute.
14. Available energy, B.T.U.
15. Theoretical water rate, pounds per kilowatt hour, B.T.U. $= \frac{44200}{\text{Avail. En. } (14)}$.
16. "Shaft" efficiency $= (15) \div (10)$.
17. "Bucket" efficiency $= (15) \div (11)$.

NOTES.—For calculating rotation loss of a new design, stage pressures are of course used.

Steam per hour is usually calculated from the area of the nozzles in the first stage if the governor is *not operating*. For a speed-torque test the *flow of steam* is constant and KW. for determining items (8) and (9) are read from this curve $\left(\text{KW.} = \frac{\text{KW.}}{\text{R.P.M.}} \times \text{R.P.M.}\right)$.

In the case of steam turbines the **net over-all efficiency** or the heat equivalent of the "shaft" kilowatts compared with the **available energy** in the Rankine or Clausius cycle is the only one of any practical value to operating engineers. It shows the engineer how his engine is working in comparison with an assumed perfect engine. A comparison of different turbines on the basis of this net over-all or "shaft" efficiency is the most satisfactory way of considering their relative merits.

Curves in **Fig. 204** are given to compare the steam consumption of a standard turbine generator and a 4-cylinder compound reciprocating steam engine of the type used by the Interurban and Metropolitan Companies of New York, assuming both

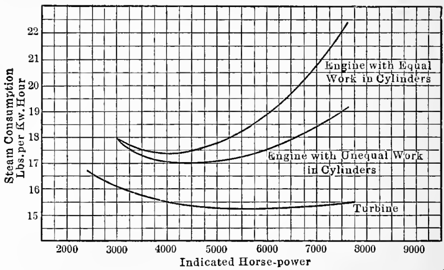


FIG. 204.—Comparative Water Rate Curves for Engines and Turbines.

units operating under the same conditions. These curves illustrate the good overload economy or the turbine, showing that at 50 per cent overload the engine designed for equal work in the cylinders requires for the same output 43 per cent more steam than the turbine.

These results are particularly interesting because the peak capacity of a station with a given equipment of boilers and auxiliaries is increased in proportion to the reduction of steam consumption at overload. For a given investment the turbine gives a much larger range of load and, moreover, affords the means by which the peak capacity of existing stations can be greatly increased.

The speed output curve (Fig. 203) is very useful to engineers to determine if a turbine is running at its best speed. If the

correspond'ng curves of steam consumption per kilowatt output (usually called water rate per kilowatt) and **efficiency** curves are calculated according to the form on page 287, a great deal of information is obtained about the operation and economy of a turbine. The **torque line** in **Fig. 203** is always drawn straight, just as a "Willans line." A curve of **total** steam consumption is usually a straight line for the normal operating limits of a turbine, but usually becomes curved when a by-pass valve opens on overload, or when the turbine is over its capacity so that the pressures are not normal in the stages.

The torque line shows why a **turbine engine is not adaptable to automobiles.** The starting torque of a small commercial turbine is not large, so that starting would be difficult with a small wheel, and reversing and speed reduction would be as difficult as with a gasoline engine. The reciprocating steam engine as well as the gasoline engine has, therefore, advantages over the steam turbine for this service.

Method of Making Tests to Determine Wheel and Blade Rotation Losses of a Steam Turbine. The simplest method for making such a test, and the one commonly employed, is to attach an electric motor to the turbine shaft (sometimes in a direct-connected set the generator is used as a motor) and run it at a number of different speeds. In taking a series of speeds, no observations are made until conditions have become "steady," and the speed must be held **constant** for several minutes so that a number of readings can be taken on the electrical instruments measuring the input of the motor. The results give the rotation loss of the wheel and blades in steam as well as bearing friction and the rotation or "windage" and electrical losses of the motor. Then the turbine wheel is removed, leaving the packing at the generator end of the turbine on the shaft, and the motor is run alone. The power now measured is that required to overcome the rotation and electrical losses of the generator and the bearing friction. Curves of power and speed as variables are plotted for each set of observations, and the disk and blade loss is determined by subtracting the ordinates of one curve from those of the other. It may be assumed with sufficient certainty that the weight of the turbine wheel itself would not alter the bearing losses to any considerable extent.[1]

[1] It may be interesting to observe that since disk and blade friction

The important fact that all results given here are for **disks and blades revolving in a stagnant medium** must not be overlooked, and it must be not assumed that the results will be the same under actual operating conditions. It may be a coincidence that the losses are the same in both cases. Under operating conditions, the spaces between the wheel blades are filled with steam flowing from the nozzle over the blades and then to the condenser. Now it has been shown by a series of experiments by Lasche of the Allgemeine Electricität Gesellschaft (Berlin) that increasing the number of nozzles around the turbine wheel reduces the disk and blade rotation losses. These losses in the blades are very largely due to the fan action of the blades which start currents of steam just as a centrifugal fan does. In other words, this is what Stodola calls "ventilation." With steam flowing through the blades, this fan action is largely prevented and the losses are consequently reduced. Another reason why the disk and blade rotation losses should be less when the turbine is operating than they are in stagnant steam is that they are really friction losses, or a conversion of kinetic energy into heat, with the effect of either superheating or drying the steam. In a turbine with more than one stage a part of the heat energy gained as the result of the friction is converted in the next expansion into kinetic energy or velocity. It is usually assumed that about 15 per cent of the disk and blade losses are regained by reheating, and that therefore the actual friction losses in an operating turbine are about this amount smaller than in stagnant steam. In cases of full admission true blade friction disappears; and a proportionate reduction will also take place, according to the degree of admission, when it is partial.

Investigation of wheel and blade friction losses by the author, using a modification of the method first suggested by Lasche of Berlin, did not show the reduction in these losses to be expected when determined under operating conditions. These results, however, cannot be considered conclusive, as the type of machine used was not well suited for the purpose, and only 25 per cent of the blades were filled with steam. It has been stated that

is proportional to the density of the medium, the friction is therefore greater in air than in dry saturated steam at atmospheric pressure. This is shown by experiments published by Lewicki in *Zeit. Verein deutscher Ingenieure*, March 28, 1903.

when a large quantity of steam passes into the casing through a suitable opening without passing through nozzles and escapes through the exhaust (without increasing the pressure), the disk and blade rotation losses are increased as much as 20 per cent. This apparently is an influence to counteract the effect of filling the blades.

In all the analysis that has preceded there are so many uncertain variables entering that it is impossible to get agreement, although, apparently, we have a large amount of data from which to draw. It may be stated, however, that all in all, the best data on disk and blade friction seem to show that it is smaller and of less significance than the results of most investigators would show.

A little space should be given to Lasche's very interesting method. A turbine-generator set was used in which the **number of nozzles discharging into the turbine could be regulated** and the output of the generator was observed for each setting of valves, and tests with varying loads were made at a number of different speeds. The turbine wheel was then removed from the shaft, and by running the generator as a motor the friction losses in the stuffing-box at the generator end of the turbine and in the bearings, as well as the windage loss of the generator, were determined. The resistance of the armature and brushes was also measured to calculate the **heating (I^2r) loss.** The sum of these losses was calculated for a number of loads (kilowatts), showing the electrical output at each speed. Another curve representing the power delivered to the shaft by the turbine was obtained by adding to the generator output for each set of nozzles open the corresponding generator losses (windage, heating, and bearing friction). The lower portions of both curves are practically straight lines, and by producing the latter curve to the horizontal axis, its intersection represents on the scale of abscissas the **disk and blade rotation losses of the turbine** at the speed of the test and **under actual operating conditions.**

By making a series of such tests at different speeds curves of rotation losses can be made. Although this method requires very careful experimenting, the same must be said of any other method of obtaining these losses. At least it must be admitted that by this method a number of *uncertain* factors to be considered in the " stagnant steam " method are eliminated.

The curves obtained by this method are really the same as "**Willans lines**" (page 274), and **might just as well be plotted for total "flow" of steam per hour as for nozzles open.** In fact in turbines where there are no nozzles the "flow" of steam must be used. It is obvious that any load curve of brake horse power giving the total steam consumption can be used to determine the rotation loss by producing the "flow" line to the axis on which the output is scaled. A good check on the results of such rotation loss tests is secured by observing whether the lines for the speeds near the rated speed cross each other at about the rated output. In a good design the speed-output curve will give nearly the same output at speeds considerably above or below the rating.[1]

The no load steam consumptions of 2000, 5000 and 9000 kilowatt **Curtis turbine generators** are respectively about 14, 12.5 and 8 per cent of that at full load. In other words these percentages are only from 1 to 2 per cent greater than the sum of the disk and blade rotation and generator windage losses. Generator windage loss is probably about equal to the sum of all the turbine losses. It is generally assumed that the no load steam consumption of a Parsons turbine (without the generator) is about 12 per cent of that at the normal **maximum output.**

It is stated[2] that at no load the steam required for very large reciprocating engines and generators is probably in no case less than 15 per cent of that used at full load.

Leakage Loss. The other important mechanical loss in a steam turbine is that due to the leakage of steam through the passages of the turbine without doing work. In impulse turbines of more than one stage this loss is chiefly caused by the leakage of steam between the shaft and the diaphragms. In a great many turbines no satisfactory packing is provided at these places and the loss is sometimes more than 10 per cent of the total anount of steam supplied to the turbine. In reaction turbines the loss is due to leakage through the radial clearance passages and is large or small in proportion to the size

[1] For a more detailed description and illustrative figures the reader is referred to the author's book on *The Steam Turbine*, pages 120–123.

[2] Kruesi, *Proc. Am. Street and Interurban Railway Engineering Association*, 1907.

of these clearances. The loss is usually assumed to be about 5 per cent in good Parsons turbines.

Future improvements in the economy of all types of steam turbines will depend largely on the success of designers in reducing these leakage losses.

Analysis of Losses. The following table shows how the losses in a De Laval 200-kilowatt turbine generator have been divided up by Stevens and Hobart:

Nozzle losses	12	per	cent
Radiation losses and leakage	1	"	"
Rotation losses due to the turbine wheel revolving in steam	4	"	"
Losses due to the steam traveling over the blades	9	"	"
Bearing friction losses	1	"	"
Losses in speed-reduction gearing	2	"	"
Generator losses	4	"	"
Losses due to residual kinetic energy in the steam passing to the condenser	8	"	"
Electrical output	59	"	"
Total	100	"	"

Electrical Output of Turbine Generators. Measurement of Direct Current. Careful engineers will not ordinarily use the instruments on the switchboard of a power station for measuring the electrical output of a generator, because, unless exceptional precautions have been taken to avoid "stray" magnetic fields and the instruments have been calibrated in place under operating conditions with a **sufficient interval of time between observations** of current (amperes) at different loads so that the **shunts of the ammeters will reach a constant temperature** for the particular value of current flowing there may be considerable error in the observations. Switchboard voltmeters are usually satisfactory if they are carefully calibrated; but the shunts of the type of ammeters ordinarily used have approximately only 60 millivolts drop, so that the indicating part of the ammeter must be almost entirely a circuit of copper wire. It is for this reason that such instruments are likely to be affected considerably by varying room temperatures, and with some shunt arrangements they are susceptible to errors, also from variations in the value of the circuit itself. For accurate measurements, it is therefore best to use only the **portable** types of indicating ammeters having shunts of 200 millivolts[1] drop. In these latter instruments the indicating part

[1] This value for the drop in shunts is an arbitrary value selected

is made up largely of resistance wires having practically no temperature coefficient. Portable voltmeters are also to be preferred to those on the switchboards.

Unless standard shunts of 200 millivolts drop as provided for good portable ammeters are used the influence of "stray" magnetic fields must be guarded against. When on the other hand switchboard instruments are used, such influences must be investigated and arrangements must be devised so that "stray" fields will not affect the measurements. The influence of very weak magnetic fields can be eliminated from the final results by turning the instruments between successive readings. Observations of current (amperes) made with the switchboard type of instruments, are also often in error due to thermoelectric effects producing a small electromotive force sufficient, however, to alter the readings of the millivoltmeter. The error due to this cause can be observed by reading the millivoltmeter at the close of the test immediately after the current has been shut off in the main circuit. It should be, of course, the object of the observer to take this reading before the shunts and leads have cooled appreciably. If there is an error due to this cause there will be a small position or negative deflection of the needle from the correct zero, which should be applied as a correction to all the observations of current.

Measurement of Alternating Current. The same general precautions outlined above for direct-current instruments, must be observed in the use of those for alternating current. Although **steady** magnetic fields are not often a cause of much trouble, it happens often, particularly in the case of large generators, that there are large magnetic fields influencing the measuring instruments which have the same frequency as that of the current measured. To eliminate the effect of such "stray" fields shielded types of instruments should be used. Only with the most expert handling can accurate results be expected when unshielded instruments are used. For measuring large values of alternating current, instrument transformers are generally used. These should be of the precision type and should be sent to a standardizing laboratory before and after a series of tests

by a number of makers of electrical instruments because it gives the best compensation of all the temperature errors. See *General Electric Review*, February, 1911.

to be calibrated, and a certificate of accuracy should be obtained. The transformers should be calibrated at as nearly as possible the values of the current to be measured in the tests.

Whenever it is possible tests of generators should be made with a non-inductive load, water rheostats being usually the most satisfactory apparatus for providing such a load. At least tests should be made under conditions giving a low power-factor, so that there can be no error in the readings of the instruments due to phase displacements in the instrument transformers. With a purely non-inductive load the readings of the ammeters and the voltmeters can be used to check the wattmeters. Although the **readings of the wattmeters should be taken as the correct value of the output** the apparent power as indicated by the ammeters and voltmeters should agree with the wattmeter readings within one per cent. If a non-inductive load cannot be secured the switchboard ammeters and voltmeters will be satisfactory for readings to indicate whether or not the load on the circuits is properly balanced. **Watt-hour meters** are not usually satisfactory for the accuracy expected in most tests, and the use of these instruments should be generally avoided. It is only in the case where tests must be made under extremely variable service conditions, where it is difficult to obtain a true average from the readings of indicating instruments, that watt-hour meter, either for direct or for alternating current, may sometimes give more accurate results than the portable indicating types of instruments. Whenever watt-hour meters are used in tests they should be checked **in place** for a series of constant loads at the frequency, voltage, etc. which are to be used in the test.

Single-phase indicating instruments are to be preferred for measurements of polyphase current to the standard types of so-called polyphase instruments. The reason for this preference is that the indications of a polyphase instrument are produced by two influences from separate phases of the current in such manner that a correction cannot be applied to obtain true values unless the division of the load is determined by the use of single-phase instruments. Obviously, then, if it is necessary to have single-phase instruments in the separate circuits, it is desirable to have them of the precision type, and polyphase instruments are not needed.

CHAPTER XIII

METHODS OF CORRECTING STEAM TURBINE AND ENGINE TESTS TO STANDARD CONDITIONS

Standard Conditions for Turbine and Engine Tests. If tests of steam turbines and engines could be always made at some **standard vacuum, superheat, and admission pressure,** then turbines and engines of the same size and of the same type could be readily compared, and an engineer could determine without any calculations which of two turbines or engines was more economical for at least these standard conditions. But steam turbines and engines even of the same make are not often designed and operated at any standard conditions, so that a direct comparison of steam consumptions has usually no significance.

It will be shown now how good comparisons of different tests can be made by a little calculation involving the reducing of the results obtained for varying conditions to **assumed standard conditions.** The method given here is that generally used by manufacturers for comparing different tests on the same turbine or engine (a " checking " process) or on different types to determine the relative performance. To illustrate the method by an application, a comparatively simple test will first be discussed.

Practical Example. Corrections for Full Load Tests. The curve in **Fig. 205** shows the steam consumption for varying loads obtained from tests of a 125-kilowatt **steam turbine** operating at 27.5 inches vacuum, 50 degrees Fahrenheit superheat, and 175 pounds per square inch absolute admission pressure (at the nozzles). It is desired to find the equivalent steam consumption at 28 inches vacuum, 0 degrees Fahrenheit superheat, and 165 pounds per square inch absolute admission pressure for comparison with the "guarantee tests" (**Fig. 206**) of a **steam engine** of about the same capacity operating at the latter

conditions of vacuum, superheat, and pressure. The manufacturers of the steam turbine have provided the curves in **Figs. 207, 208, and 209**, showing the change of economy with

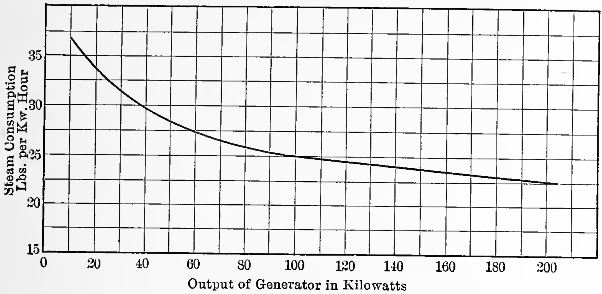


FIG. 205.—Water Rate Curve of a Typical 125-Kilowatt Steam Turbine. (Generator Output.)

varying vacuum, superheat, and pressure. With the help of these **correction curves**, the steam consumption of the turbine can be reduced to the conditions of the engine tests. **Fig.**

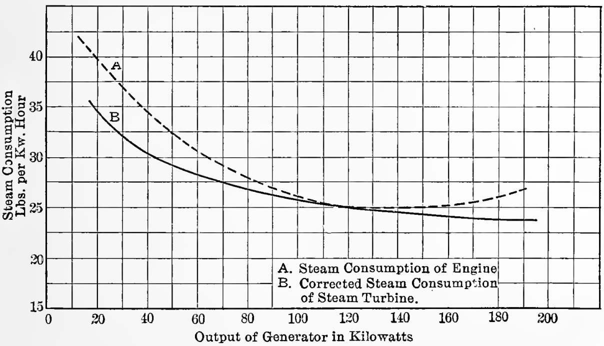


FIG. 206.—Comparative Water Rate Curves of a Reciprocating Steam Engine and a Steam Turbine. (Both with Standard Generators.)

207 shows that between 27 and 28 inches vacuum a difference of 1 inch changes the steam consumption 1.0 pound. **Fig. 208** shows a change of 2.0 pounds per 100 degrees Fahrenheit

superheat, and from **Fig. 209** we observe a change of 5.0 pounds in the steam consumption for 100 pounds difference in admission pressure. Compared with the engine tests the steam turbine was operated at .5 inch lower vacuum, 50 degrees Fahrenheit higher superheat, and 10 pounds higher pressure. At the con-

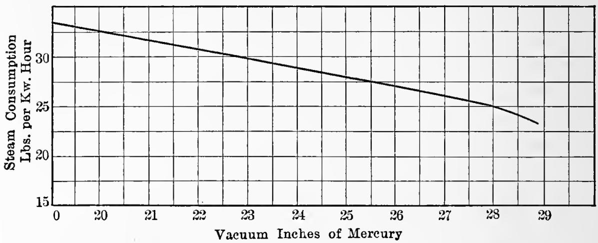


FIG. 207.—Vacuum Correction Curve for a 125-Kilowatt Steam Turbine.

ditions of the engine tests, then, the steam consumption of the steam turbine should be reduced .5 pound to give the equivalent at 28 inches vacuum, but is increased 1.0 pound to correspond to 0 degrees Fahrenheit superheat, and .5 pound more to bring it to 165 pounds absolute admission pressure. The full load

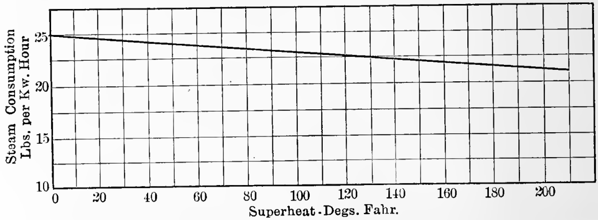


FIG. 208.—Superheat Correction Curve for a 125-Kilowatt Steam Turbine.

steam consumption for the steam turbine at the conditions required for the comparison is, therefore, $24.5 - .5 + 1.0 + .5$, or 25.5 pounds.[1]

[1] The *corrected* steam consumption is found to be nearly the same as that which the three correction curves show for the same conditions, that is, about 25.0 pounds. If there had been a difference of more than about 5 per cent between the corrected steam consumption and

Persons who are not very familiar with the method of making these corrections will be liable to make mistakes by not knowing whether a correction is to be added or subtracted. A little thinking before writing down the result should, however, prevent such errors. When the performance at a given vacuum is to be **corrected to a condition of higher vacuum, the correction must be subtracted,** because obviously the steam consumption is reduced by operating at a higher vacuum. When the steam consumption with superheated steam is to be determined in its equivalent of dry saturated steam (0 degrees superheat) the correction must be added because with lower superheat there is less heat energy in the steam and consequently there is a

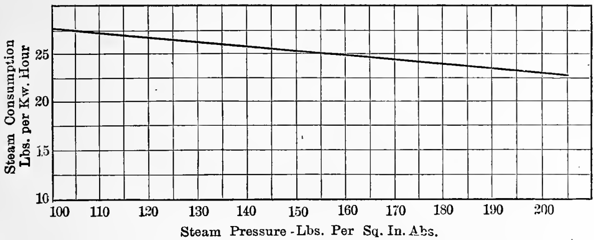


FIG. 209.—Pressure Correction Curve for a 125-Kilowatt Steam Turbine.

larger consumption. Usual corrections for differences in admission pressure are not large; but it is well established that the economy is improved by increasing the pressure.

Corrections for Fractional Loads. It is the general experience of steam turbine manufacturers that **full-load correction curves,** if used by the following "ratio" or percentage method, can be used for correcting **fractional** or **over loads.** This statement applies at least without appreciable error from half to one and a half load, and is the only practicable method for quarter load as well.[1] Stated in a few words, it is assumed then that the steam consumption at fractional loads is changed by the same percentage as at full load for an inch of vacuum,

that of the correction curves for the same conditions, the "ratio" method as explained on page 300 for fractional loads should have been used also for full load.

[1] A very exhaustive investigation of this has been made by T. Stevens and H. M. Hobart, which is reported in *Engineering*, March 2, 1906.

a degree of superheat, or a pound pressure. It will now be shown how this method applies to the correction of the steam consumption of the turbine at fractional loads. Now according to the curve in **Fig. 207,** the steam consumption at 27.5 inches (25.6 pounds) must obviously be multiplied by the ratio[1] $\frac{25.0}{25.6}$, of which the numerator is the steam consumption at 28 inches and the denominator at 27.5 inches, to get the equivalent consumption at 28 inches vacuum. This reasoning establishes the proper method for making corrections; that is, that the **base for the percentage** (denominator of the fraction) must be the steam consumption at the condition to which the correction is to be applied.[2] Similarly the correction ratio to change the consumption at 50 degrees Fahrenheit superheat to 0 degrees Fahrenheit is $\frac{25.0}{24.0}$, and to correct 175 pounds pressure to 165 pounds the ratio is $\frac{24.8}{24.3}$. Data and calculated results obtained by this method may then be tabulated as follows:

	Conditions of Test.	Required Conditions.	Correction Ratio.	Percentage Correction.
Vacuum, inches	27.5	28	$\frac{25.0}{25.6}$	−2.34%*
Superheat, degrees Fahrenheit	50.	0	$\frac{25.0}{24.0}$	+4.17%
Admission pressure, pounds absolute	175.	165	$\frac{24.8}{24.3}$	+2.06%
Net correction	....	....	....	+3.89%

* Steps in the calculation are omitted in the table, thus $\frac{25.0}{25.6}=.9766$ or 97.66 per cent making the correction 100−97.66, or 2.34 per cent. It may seem unreasonable to the reader that these percentages are calculated to three figures when the third figure of the values of steam consumption is doubtful. In practice, however, the ruling of the curve sheets must be much finer and to larger scale so that the curves can be read more accurately.

[1] Assuming that this short length of the curve may be taken for a straight line without appreciable error.

[2] In nearly all books touching this subject so important to the practical, consulting, or sales engineer, the alternative method of taking the steam consumption at the required conditions as the base for the percentage calculation is implied. By such a method percentage correction curves derived from straight lines like Figs. 208 and 209 would be straight lines and, in application, give absurd results. Actually such percentage corrections will fall on curves.

The signs + and − are used in the percentage column to indicate whether the correction will increase or decrease the steam consumption. "Net correction" is the algebraic sum of the quantities in the last column.

The following table gives the results of applying the above "net correction" to fractional loads.

	¼ Load 31.3 kw.	½ Load 62.5 kw.	¾ Load 93.8 kw.	4/4 Load 125 kw.	5/4 Load 156.3 kw.
Steam consumption from test (Fig. 205)...........	31.2	26.9	25.2	24.5	23.6
Net correction +3.89%.....	+1.2	+1.1	+1.0	+1.0	+0.9
Corrected steam consumption	32.4	28.0	26.2	25.5	24.5

Curve *B* in **Fig. 206** shows the corrected curve of steam consumption for the steam turbine as plotted from the above table. By thus combining, on the same curve sheet, curves *A* and *B* as in this figure, the points of better economy of the turbine are readily understood.

Results of economy tests of the various turbines given on the preceding pages are of very little value for comparison when the steam consumptions or "water rates" are given for all sorts of conditions. With the assistance, however, of curves like those shown in **Figs. 207, 208** and **209**, if they are representative of the type and size of turbine tested, it is possible to make valuable comparisons between two or more different turbines. Some very recent data of Curtis and Westinghouse-Parsons turbines are given below, together with suitable corrections adopted by the manufacturers for similar machines.

The following test of a Westinghouse-Parsons turbine, rated at 7500 kilowatts, was taken at Waterside Station No. 2 of the New York Edison Co., and a comparison is made with a test of a five-stage 9000-kilowatt Curtis turbine at the Fisk Street Station of the Commonwealth Electric Company of Chicago. As no pressure correction is given for the Curtis machine, the New York Edison test is corrected to the pressure at which the other machine was operated (179 pounds per square inch gage). Approximately an average vacuum for the two tests is taken for the standard, and 100 degrees Fahrenheit superheat is used for comparing the superheats. These assumed standard con-

7500 KILOWATT WESTINGHOUSE-PARSONS TURBINE, WATERSIDE STATION NO. 2, NEW YORK EDISON COMPANY

Tested September 1, 1907

		Corrected to	Correction per cent.*
Duration of test, hours	8	....	....
Speed revolutions per minute	750	....	....
Average steam pressure, pounds gage	177.5	179	− .15
Average vacuum, ins. (referred to 30 in. barom.)	27.3	28.5	−3.36
Average superheat, degrees Fahrenheit	95.7	100	−0.29
Average load on generator, kilowatts	9830.5	....	....
Steam consumption, pounds per kilowatt-hour	15.15†	....	....
Net correction, per cent	....	....	−3.80
Corrected steam consumption, pounds per kilowatt-hour	....	14.57	....

* The following corrections were given by the manufacturers and accepted by the purchaser as representative of this type and size of turbine:

Pressure correction .1 per cent for 1 pound.
Vacuum correction 3.5 per cent for 1 inch.
Superheat correction 7.0 per cent for 100 degrees Fahrenheit.
} from *Electric Journal*, November, 1907, page 658.

† This is $7\frac{1}{2}$ per cent better than the manufacturer's guarantee.

9000 KILOWATT CURTIS TURBINE, FISK STREET STATION, COMMONWEALTH ELECTRIC COMPANY, CHICAGO.

Tested in 1907

		Corrected to	Correction, per cent.*
Duration of test	....	....	....
Speed, revolutions per min	750	....	....
Average steam pressure, pounds gage	179	179	.0
Average vacuum, inches (referred to 30 in. barom.)	29.55	28.5	+12.39
Average superheat, degrees F	116	100	+ 1.28
Average load on generator, kilowatts	8070.	....	....
Steam consumption, pounds per kilowatt-hour	13.0	....	....
Net correction, per cent	....	....	+13.67
Corrected steam consumption, pounds per kilowatt-hour	....	14.77	....

* The following percentage corrections were used:

Superheat corrections 8 per cent for 100 degrees Fahrenheit.
Vacuum correction 8 per cent for 1 inch from curve in Fig. 210.
Pressure correction not given.
} *G. E. Bulletin*, No. 4531.

ditions make the corrections for each turbine comparatively small. When two tests are to be compared, by far the more intelligent results are obtained if each is corrected to the average conditions of the two tests, rather than correcting one test to the conditions of the other. There is always a chance for various errors when large corrections must be made.

These results show a difference of only .20 pound in the corrected steam consumption, so that for exactly the same conditions these two machines would probably give approximately the same economy. Each turbine is doubtless best for the special conditions for which it was designed.

These results are *equivalent* to respectively 9.58 pounds and 9.72 pounds per **indicated horse power,** assuming 97 per cent as the efficiency of the generator and 91 per cent as the mechanical efficiency of a large Corliss engine according to figures given by Stott.[1]

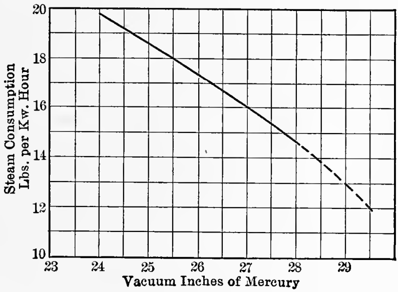


FIG. 210.—Typical Vacuum Correction Curve for a 5000-Kilowatt Impulse Turbine.

From experience with other similar turbines it seems as if the vacuum corrections given are too low for each turbine. The correction for the Curtis turbine was obtained from the curve in **Fig. 210,** as given between 27 and 28 inches, while it was used between 28.5 and 29.5 inches, where the curve of steam consumption most likely slopes somewhat, as shown by the dotted line in the figure, which was derived from the percentage change of the theoretical steam consumption calculated from the available energy. The correction of 2.7 per cent per inch of vacuum for the Westinghouse-Parsons turbine is probably too low also, although the percentage correction

[1] *Electric Journal,* July, 1907. It is stated also in this article that the vacuum correction of a Westinghouse-Parsons turbine is 3.5 per cent per inch between 28 and 28.5 inches. Jude states that the vacuum correction for Parsons turbines is 5 to 6 per cent.

would not be nearly as large as for the Curtis. If both of these corrections are too low, the effect of increasing them would be to increase the corrected steam consumption of the Curtis turbine and reduce that of the Westinghouse-Parsons.

Tests of a 5000-kilowatt Curtis and a 7500-kilowatt Westinghouse-Parsons turbine are also recorded here for comparison. The two tests are corrected to the assumed standard conditions of 173.7 pounds gage pressure, 28 inches vacuum, and 0 degrees Fahrenheit superheat. For the test of the Curtis machine the same percentage corrections were used as for the 9000-kilowatt turbine; and for the test of the Westinghouse turbine the vacuum correction is that given in the footnote at the middle of page 302 (3.5 per cent per inch), while the other percentage corrections are the same as in the preceding test of a similar machine. The Westinghouse turbine was operated with wet steam. In a test of a reciprocating engine the equivalent economy with dry steam is calculated by merely subtracting the percentage of moisture, but **in a turbine test the correction is generally stated as being a little more than twice the percentage of moisture.** In other words, in a turbine test the moisture must be subtracted twice. The reason for this difference in the methods of correcting water rates of engines and turbines is the very large increase in the disk and blade rotation losses in wet steam

5000-KILOWATT FIVE-STAGE CURTIS TURBINE, L STREET STATION, BOSTON EDISON COMPANY

Tested January 29, 1907

		Corrected to	Correction, per cent.
Duration of test, hours	3	....	....
Speed, revolutions per minute	720	....	....
Average steam pressure, pounds gage	173.7	173.7	0
Average vacuum in. (referred to 30 in. barom.)	28.8	28	+ 6.40
Average superheat, degrees Fahrenheit	142	0	+11.36
Average load on generators, kilowatts	5195	....	....
Steam consumption, pounds per kilowatt-hour	13.52	....	....
Net correction	....	....	+17.76
Corrected steam consumption, pounds per kilowatt-hour	....	15.92	....

7500-KILOWATT WESTINGHOUSE-PARSONS TURBINE. INTERBOROUGH RAPID TRANSIT COMPANY, NEW YORK

Tested in 1907

		Corrected to.	Correction, per cent.
Duration of test, hours	3	....	....
Speed, revolutions per minute	....	....	....
Average steam pressure, pounds gage	149.7	173.7	−2.4
Average vacuum, in. (referred to 30 in. barom.)	27.70	28	−1.05
Average moisture, per cent	3.0	0	−6.0
Average load on generator, kilowatts	7135	....	....
Steam consumption, pounds per kilowatt-hour (wet)	17.79	....	....
Net correction	....	....	−9.45
Corrected steam consumption, pounds per kilowatt-hour	....	16.10	....

It is stated that the steam consumption of the Interborough Company's turbine is 15.87 pounds at full load and 15.54 pounds at 9000 kilowatts when the overload valve opens. The generator connected to this turbine is rated at only 5500 kilowatts. With a generator more nearly the rating of the turbine it is probable still better results would be secured.

CHAPTER XIV

GAS AND OIL ENGINE AND PRODUCER TESTING

THE testing of internal combustion engines of the reciprocating type operating with gas, gasoline, kerosene, and alcohol does not differ essentially in the important details from steam engine practice already explained in Chapter XI. Indicator diagrams must be utilized to show the inner workings in the engine cylinder, giving a record of the pressure, "timing" of the valves and ignition for the operation of the engine through a cycle.[1]

Brake horse power is measured with a Prony brake or any other type of dynamometer permitting the determination with facility of the power of the engine. If with a Prony brake or similar device (see page 122) then the **brake horse power** is expressed by the usual formula,

$$\text{B.H.P.} = \frac{2\pi lnw}{33,000} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (87)$$

where **l** is the length of the brake-arm in **feet, n** is the number of revolutions per minute, and **w** is the **net** weight indicated by the scales on the brake. Similarly the **indicated horse power** is given by the usual formula for a single-acting steam engine (page 118) except that the **number of explosions** must be used in calculations instead of the number of revolutions, thus,

$$\text{I.H.P.} = \frac{plan_e}{33,000} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (88)$$

[1] In what is called usually a four-cycle engine there are four piston strokes—one each for suction, compression, expansion, and exhaust, corresponding to two revolutions of the crank shaft for a complete cycle, while in a so-called two-cycle engine, two strokes of the piston make a complete cycle—corresponding to one revolution of the crank-shaft. In the latter case suction and compression are combined in one stroke and expansion and exhaust in another.

where **p** is the mean effective pressure in pounds per square inch measured from the indicator diagram, **l** is the length of the stroke in feet, **a** is the area of the piston in square inches, and $\mathbf{n}_e$ is the number of explosions per minute, then

$$\textbf{Mechanical Efficiency} = \frac{\textbf{B.H.P.}}{\textbf{I.H.P.}} \quad . \quad . \quad . \quad (89)$$

Certain important precautions should be observed when taking indicator diagrams, so that a reasonable degree of accuracy may be expected from the results of tests. In the first place the connections between the indicator and the combustion chamber should be as short as possible. It is much more important that in an internal-combustion engine the volume by which the combustion chamber (clearance) is increased by the indicator connections should be small in comparison to the volume of the engine cylinder than in a steam engine; because by increasing the clearance volume, obviously, the pressure resulting from compression is reduced as well as the pressure due to the explosion. It is in this way that large indicators and indicator connections may cause a considerable reduction in the thermal efficiency of an engine, that is, reducing the efficiency of the transformation of heat energy into work.

Tests of gas engines as made commercially have usually three objects in view:

(1) Brake horse power.

(2) Indicated horse power.

(3) Gas or oil consumption per horse power per hour.

Many types of gasoline engines, particularly those designed for the automobile service, operate at such high speeds that the indicated horse power cannot be obtained with accuracy. In many other kinds of engines classed in this group the mechanical efficiency is very low. It is for these reasons that such engines are rated by the useful or brake horse power instead of by the indicated horse power, as with steam engines. Brake horse power is therefore the prime criterion by which the performance of these engines is expressed.

Ordinary types of steam engine indicators, moreover, are not very satisfactory for testing gas engines, and many engineers prefer to use one of the type shown in the accompanying illustration, **Fig. 211.** It differs essentially from steam engine

indicators of the same type in having in the lower part of the main "barrel," a cylinder of smaller diameter than the one just above it, containing the spring. This smaller cylinder takes a piston of only half the area of the standard size. By this means the scale of the spring ordinarily used is doubled and the shock on the small rods and levers of the pencil motion

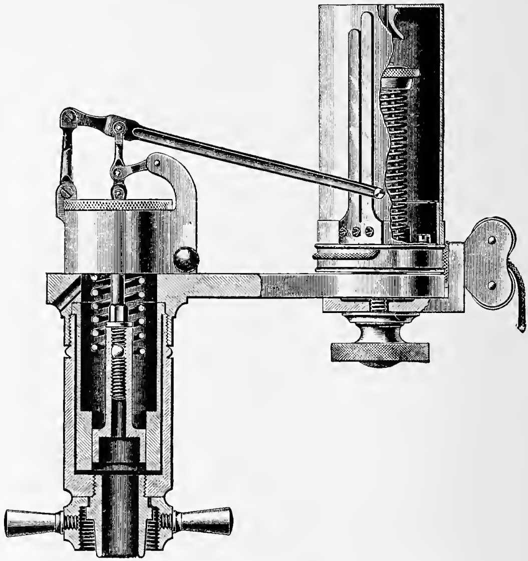


FIG. 211.—Crosby Gas Engine Indicator.

is only half as great, thus making the liability to breakage and the cost of repairs for such indicators very much less than when a piston of the standard size is used.

Measurement of the Fuel Used. For tests of gas engines the gas used is usually measured by means of a gas meter. A so-called "wet" meter (page 143) is always to be preferred, but if a carefully calibrated dry meter is used very good results can be obtained and it is accurate enough for commercial tests. For gasoline, kerosene, and other oil engines the amount of fuel used is preferably determined by direct weighing. The

author has found the automatic indicating scales of the **pendulum** type [1] now generally used by grocers and meat dealers to be most satisfactory. By this means the weight of the oil remaining in the "supply" vessel can be observed regularly throughout a test just as with a gas meter, so that irregularities in operation can be immediately observed. The vessel containing the oil used by the engine when placed on a scales must be connected to the carbureter or the pump, as the case may be, with a very flexible metallic tube made without rubber insertion.[2] If an indicating scales is not available a very small platform or grocers beam scales can be used satisfactorily, although if the weight of fuel oil is desired at regular intervals throughout a test some little time is required to balance the poise.

Another method very commonly used, however, is to use a cylinder vessel of small diameter provided with a gage glass in which the level of the liquid can be observed. Such a vessel can be calibrated to determine the weight or volume of oil per inch of height measured on the gage glass.

Observations taken for a test of a gas or oil engine are in general much more uniform than the corresponding data taken in a steam engine trial. For this reason gas engine tests in particular can be made of much shorter duration than upon a steam engine for the same degree of accuracy.

For both gas and oil engines the points plotted on a scale of brake horse power for abscissas and fuel used per unit of time for ordinates will fall along a straight line, similar and resembling the **Willans line** for steam engines and steam turbines (see page 274). A typical set of curves of the results of a test of a gas engine is shown in **Fig. 212.** Brake horse power is taken for the abscissas, as should always be done for gas and oil engine tests and gas used per hour, to its corresponding scale of ordinates, is given by the top curve. Other curves show the gas used per brake horse power per hour and per indicated

[1] Very satisfactory scales for this purpose are made by the Toledo Scale Co. of Toledo, Ohio. Similar scales of the **spring** type are not recommended, because necessarily some of the vibrations of the engine will be transmitted to the scales and the indications of the pointer will not be as accurate as they should be.

[2] Tubes of this kind are made by the U. S. Metallic Tube Co. of Los Angeles, Cal.

horse power per hour, the indicated horse power, the **mechanical efficiency,** the **number of explosions** per minute, the **revolutions per minute** and the **thermal efficiency** (heat equivalent of the indicated horse power divided by heat supplied).

The following paragraphs and tabular forms are taken from the "Rules for Conducting Tests of Gas or Oil Engines" as adopted by the American Society of Mechanical Engineers.[1]

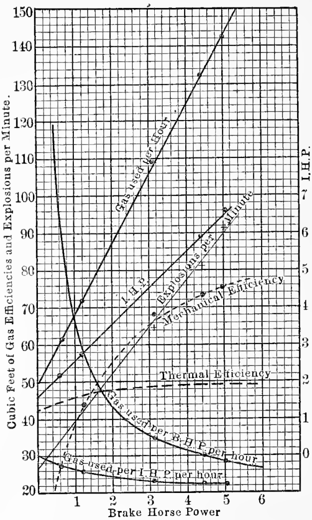


FIG. 212.—Typical Economy, Speed, Horse Power and Efficiency Curves of a Five Horse Power Gas Engine.

Duration of Test. The duration of a test should depend largely upon its character and the objects in view, and in any case the test should be continued until the successive readings of the rates at which the oil or gas is consumed, taken at, say, half-hourly intervals, become uniform and thus verify each other. If the object is to determine the working economy, and the period of time during which the engine is usually in motion is some part of twenty-four hours, the duration of the test should be fixed for this number of hours. If the engine is one using coal for generating the gas, the test should cover a long enough period of time to determine with accuracy the coal used in the **gas producer;** such a test should be of at least twenty-four hours' duration, and in **most cases it should extend over several days.**

Measurement of Fuel. If the fuel used is coal furnished to a gas producer, the same methods apply for determining the consumption as are used in steam boiler tests. (See pages 210–217.)

[1] *Transactions American Society of Mechanical Engineers.* Vol. 24, pages 775–790.

If the fuel used be gas, the only practical method of measuring is the use of a meter through which the gas is passed. Gas bags should be placed between the meter and the engine to diminish the variations of pressure, and these should be of a size proportionate to the quantity used. Where a meter is employed to measure the air used by an engine, a receiver with a flexible diaphragm should be placed between the engine and the meter. The temperature and pressure of the gas should be measured, as also the barometric pressure and temperature of the atmosphere.

Data and Results of test of Gas or Oil Engine

Standard Form

1. Made by .... of ....
   on engine located at ....
   to determine ....
2. Date of trial ....
3. Type of engine, whether oil or gas ....
4. Class of engine (mill, marine, motor, for vehicle, pumping or other) ....
5. Number of revolutions for one cycle and class of cycle ....
6. Method of ignition ....
7. Name of builders ....
8. Gas or oil used ....
   (*a*) Specific gravity ....
   (*b*) Flashing point, deg. F.[1] (open or closed vessel) ....
   (*c*) Burning point, deg. F. ....
9. Dimensions of engine ....
   (*a*) Class of cylinder (working or for compressing the charge) .....
   (*b*) Vertical or horizontal ....
   (*c*) Single or double acting ....
   (*d*) Cylinder dimensions ....
   Bore, ins. ....
   Stroke, ft. ....
   Diameter piston rod, ins. ....
   Diameter tail rod, ins ....
   (*e*) Compression space or clearance in per cent of volume displaced by piston per stroke ....
   Head end ....
   Crank end ....
   Average ....

[1] The *flashing-point* of fuel oils lighter than about 70° Baumé (0.70 specific gravity) is at "room" temperatures so that no test is needed. The flashing-point of heavier oils is usually made by heating the oil in a vessel closed at the top with a glass plate in which there are openings for the insertion of a thermometer and for testing with a lighted match or taper. The flash-point is the temperature observed when the vapor on the surface of the oil first flashes. If the flashing-point is determined in an open vessel the value is considerably lower than in a closed vessel as described above. To determine the ***burning point*** the glass cover is to be removed and the heating continued till the whole surface of the oil takes five and must be blown out. The flame should be extinguished quickly so that it will not unduly heat the thermometer and raise the temperature to be observed as the burning point.

(f) Surface in sq. ft. (average) ....
Barrel of cylinders ....
Cylinder heads ....
Clearance and ports ....
End of piston ....
Piston rod ....

(g) Jacket surfaces or internal surfaces of cylinder heated by jackets in sq. ft. ....
Barrel of cylinder ....
Cylinder heads ....
Clearance and ports ....

(h) Horse power constant for one lb. M.E.P. and one revolution per minute ....

10. Give description of main features of engine and plant and illustrate with drawings of same given on an appended sheet. Describe method of governing. State whether the conditions were constant throughout the test ....
....
....

Total Quantities.

11. Duration of test, hours ....
12. Gas or oil consumed, cu. ft. or lbs. ....
13. Air supplied in cubic feet ....
14. Cooling water supplied to jackets, lbs. ....
15. Calorific value of gas or oil by calorimeter test, determined by ....calorimeter, B.T.U. ....

Hourly Quantities.

16. Gas or oil consumed per hour, cu. ft. or lbs. ....
17. Cooling water supplied per hour, lbs. ....

Pressures and Temperatures.

18. Pressure at meter (for gas engine) in inches of water ....
19. Barometric pressure of atmosphere ....
(a) Reading of height of barometer, ins. ....
(b) Reading of temperature of barometer, deg. F. ....
(c) Reading of barometer corrected to 32° F., ins. ....
20. Temperature of cooling water ....
(a) Inlet, deg. F. ....
(b) Outlet " ....
21. Temperature of gas at meter (for gas engine), deg. F. ....
22. Temperature of atmosphere, deg. F. ....
(a) Dry-bulb thermometer " ....
(b) Wet-bulb thermometer " ....
(c) Degree of humidity, per cent, (see page 331) ....
23. Temperature of exhaust gases, deg. F. ....
How determined ....

Data Relating to Heat Measurements.

24. Heat units consumed per hour (lbs. of oil or cu. ft. of gas per hour multiplied by the total heat of combustion), B.T.U. ....

25. Heat rejected in cooling water..............................
 (a) Total per hour, B.T.U..............................
 (b) In per cent of heat of combustion of the gas or oil consumed..............................
26. Sensible heat rejected in exhaust gases above temperature of inlet air..............................
 (a) Total per hour, B.T.U..............................
 (b) In per cent of heat of combustion of gas or oil consumed.....
27. Heat lost through incomplete combustion and radiation per hour....
 (a) Total per hour, B.T.U..............................
 (b) In per cent of heat of combustion of the gas or oil consumed...

Speed, etc.

28. Revolutions per minute..............................
29. Average number of explosions per minute..............How determined..............................
30. Variation of speed between no load and full load..................
31. Fluctuation of speed on changing from no load to full load measured by the increase in the revolutions due to the change.........

Indicator Diagrams.

32. Pressure in lbs. per sq. in. above atmosphere, 1 cyl...............
 2 cyl..............................
 (a) Maximum pressure..............................
 (b) Pressure just before ignition..............................
 (c) Pressure at end of expansion..............................
 (d) Exhaust pressure..............................
 (e) Mean effective pressure..............................
33. Temperatures in degrees F. computed from diagrams.............
 (a) Maximum temperature (not necessarily at maximum press.) ..
 (b) Just before ignition..............................
 (c) At end of expansion..............................
 (d) During exhaust..............................
34. Mean effective pressure in lbs. per sq. in..............................

Power.

35. Power as rated by builders..............................
 (a) Indicated horse power..............................
 (b) Brake horse power..............................
36. Horse power (indicated) actually developed....................
 First cylinder..............................
 Second cylinder..............................
 Total..............................
37. Brake horse power, electric horse power or pump horse power according to the class of the engine......................
38. Friction indicated horse power from diagrams with no load on engine and computed for average speed...................
39. Percentage of indicated horse power lost in friction...............

Standard Efficiency Results.

40. Heat units consumed by the engine per hour, B.T.U..............
 (a) Per indicated horse power..............................
 (b) Per brake horse power..............................

41. Heat units consumed by the engine per minute, B.T.U. . . . .
(*a*) Per indicated horse power . . . .
(*b*) Per brake horse power . . . .
42. Thermal efficiency, ratio per cent . . . .
(*a*) Per indicated horse power . . . .
(*b*) Per brake horse power . . . .

Miscellaneous Efficiency Results.

43. Cubic feet of gas or lbs. of oil consumed per horse power per hour. . . .
(*a*) Per indicated horse power . . . .
(*b*) Per brake horse power . . . .

Heat Balance.

44. Quantities given per cent of the total heat of combustion of the fuel. . . . .
(*a*) Heat equivalent of indicated horse power. . . . .
(*b*) Heat rejected in cooling water . . . .
(*c*) Heat rejected in exhaust gases and lost through radiation and incomplete combustion . . . .
Subdivisions of Item (*e*)
(1) Heat rejected in exhaust gases . . . .
(2) Heat lost through incomplete combustion. . . . .
(3) Heat lost through radiation and unaccounted . . . .

Data and Results of Standard Heat Test of Gas or oil Engine.

Short Form

1. Made by. . . . . .of. . . . .
on engine located at. . . . .
to determine. . . . .
2. Date of trial. . . . .
3. Type of engine or class . . . .
4. Kind of fuel used . . . .
(*a*) Specific gravity . . . .
(*b*) Flashing point, deg. F. . . . .
(*c*) Burning point " . . . .
5. Dimensions of engines . . . .
(*a*) Class of cylinder (working or for compressing the charge) . . . . .
(*b*) Single or double acting. . . . .
(*c*) Cylinder dimensions . . . .
Bore. . . . . . . .ins. . . . .
Stroke. . . . . .ft. . . . .
Diameter of piston ins. . . . .
(*d*) Average compression space, or clearance in per cent of volume displaced in piston per stroke. . . . .
(*e*) Horse power constant for one pound M.E.P. and one revolution per minute . . . .

Total Quantities.

6. Duration of test, hours . . . .
7. Gas or oil consumed, cu. ft. or lbs. . . . .
8. Cooling water supplied to the jackets, lbs. . . . .
9. Calorific value of fuel by calorimeter test, determined by. . . . . . calorimeter, B.T.U. . . . .

Pressures and Temperatures.

10. Pressure at meter (for gas engine) in inches of water..............
11. Barometric pressure of atmosphere..............................
    (*a*) Reading of barometer in inches..........................
    (*b*) Reading corrected to 32° F, ins..........................
12. Temperature of cooling water, degrees F........................
    (*a*) Inlet...
    (*b*) Outlet..
13. Temperature of gas at meter (for gas engine) degrees F............
14. Temperature of atmosphere, degrees F...........................
    (*a*) Dry-bulb thermometer, degrees F........................
    (*b*) Wet-bulb thermometer, " ....
    (*c*) Degree of humidity, per cent............................
15. Temperature of exhaust gases, deg. F..........................

Data Relating to Heat Measurements.

16. Heat units consumed per hour (pounds of oil or cubic feet of gas per hour multiplied by the total heat of combustion), B.T.U.......
17. Heat rejected in cooling water per hour, B.T.U...................

Speed, etc.

18. Revolutions per minute.......................................
19. Average number of explosions per minute.......................

Indicator Diagrams.

20. Pressure in lbs. per sq. in. above atmosphere....................
    (*a*) Maximum pressure..
    (*b*) Pressure just before ignition............................
    (*c*) Pressure at end of expansion............................
    (*d*) Exhaust pressure..
20a. Mean effective pressure, lbs. per sq. in.........................

Power.

21. Indicated horse power..
    First cylinder..
    Second cylinder...
    Total...
22. Brake horse power...
23. Friction horse power by friction diagram........................
24. Percentage of indicated horse power lost in friction...............

Standard Efficiency and Other Results.

25. Heat units consumed by the engine per hour, B.T.U...............
    (*a*) Per indicated horse power...............................
    (*b*) Per brake horse power...................................
26. Pounds of oil or cubic feet of gas consumed per hour..............
    (*a*) Per indicated horse power, lbs. or cu. ft....................
    (*b*) Per brake horse power, lbs. or cu. ft.......................

Indicator Diagrams of the Suction Stroke of a Gas or Oil Engine. With the ordinary stiff spring used for measuring

the horse power of gas and oil engines, very little information regarding the action of the valves during the suction stroke is obtainable from the indicator diagram. For this reason the events in the suction stroke must be obtained with a comparatively light spring, which must be protected, however, from injury when subjected to the excessive pressure of the explosion stroke by inserting a suitable stop of some kind to

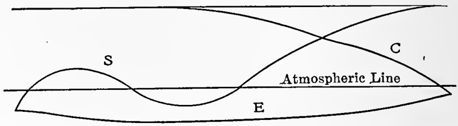


FIG. 213.—" Light Spring " Indicator Diagram of a Gas Engine.

prevent undue compression of the spring. The device usually adopted is to slip a small brass tube over the piston rod of the indicator to act as a distance piece. Another method, also satisfactory, is to fit a short but very thin brass tube over the outside of the spring, but of such a diameter that it will pass easily inside the cylinder of the indicator and rest easily on the top of the piston. A " light-spring " diagram is shown in **Fig.**

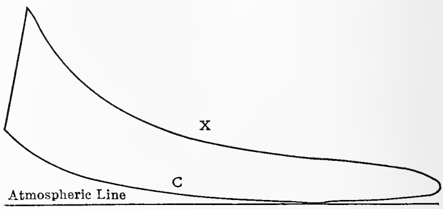


FIG. 214.—Normal Indicator Diagram of a Gas Engine.

213, which was taken from an engine giving an ordinary diagram like **Fig. 214.** In **Fig. 213** the lower horizontal line is the atmospheric line and the upper horizontal line is traced by the pencil of the indicator, during the compression and explosion strokes, showing the effect of the stop. The wavy line **S** shows the **exhaust** stroke, and the slightly curved line **E** is the **suction.** The diagram shows that there was a partial vacuum **throughout** the suction stroke and for a part of the exhaust stroke, the

the latter effect being due doubtless to the inertia of the gases in the exhaust pipe.

In the three figures following very interesting indicator diagrams of gas engines due to Pullen are illustrated. **Figs. 215** and **216** show explosions during the suction stroke, generally called explosions in the air pipe, for the reason that since the air valve is then open the explosion occurred probably in the air pipe. In **Fig. 215** the effect of the explosion is shown in the indicator diagram by the hump near the atmospheric line near the middle of the stroke, while in **Fig. 216** the explosion

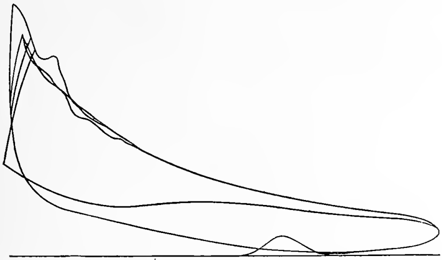


FIG. 215.—Indicator Diagram of a Gas Engine Showing Explosion in Air Pipe.

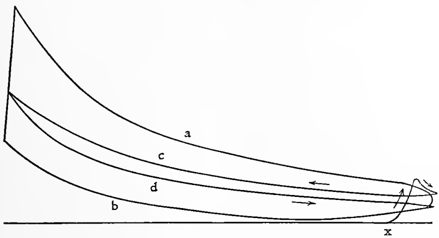


FIG. 216.—Abnormal Gas Engine Diagrams.

occurred near the end of the stroke. Explosions in the air pipe are sometimes attributed to there being too weak a mixture (too little rich gas) in the cylinder, causing very slow burning instead of a sharp explosion. Under these conditions combustion will not be complete at the end of the working stroke, and

this slow burning goes on through the exhaust stroke. Then when the exhaust valve closes, some of this smouldering gas remains in the clearance space, which, when mixed with the new charge during the next suction stroke, forms a combustible mixture which is easily exploded. In **Fig. 216** the explosion occurred near the end of the **suction** stroke at **x**, and the air valve has closed before the pressure has had time to fall to atmospheric. On this account the compression line **c** is much higher than it would be under normal conditions as shown by **b.** Since no explosion takes place, the curve **d** corresponding to the working stroke lies just below this abnormal compression line.

No less interesting are the diagrams illustrated in **Fig. 217,** showing the effect of pre-ignition on the indicator diagram of

Fig. 217.—Indicator Diagrams of a Gas Engine Illustrating the Effect of "Timing" (from Preignition to Slow Burning).

a gas engine. Here in two of the diagrams shown the ignition occurred too early or before the end of the compression stroke. Under these conditions there is usually a heavy thumping noise in the engine cylinder, and the engine will not develop as much power as there would be if ignition were "timed" a little later. This effect may be caused in poorly designed engines by some small metal projection or web in the clearance space becoming hot enough to ignite the charge before the ignition device operates. On the other hand, the point of ignition may have been advanced too far by inexperienced persons.

Fuels for Gas and Oil Engines. The ordinary type of gas engine is generally operated with either **illuminating gas** or **natural gas.** Since, however, natural gas occurs only in limited

areas its use is very much restricted. **Blast-furnace gas** is used in iron works for operating engines with the waste gases from the blast furnaces. The **gas from coke-ovens,** also a waste gas, is now being used to some extent in the gas engines of power plants in the coke regions. Of the various kinds of so-called fuel gases, **producer gas** is, however, by far the most important. Anthracite coal is more easily converted into producer gas than any other fuel, although cheap bituminous coals are now also used. The apparatus used for generating producer gas is called in technical language a **producer.** There are in common use two types of producers for converting solid fuel into a permanent fuel gas. In one type the air or the steam (or both together) that is required for the operation of the producer is forced under pressure, produced usually by a blower, through the bed of solid fuel. In the other type of producer the air and water are drawn through the bed of fuel either by the **suction** of the engine itself, or by the suction of an auxiliary "exhauster." Gas is made at a more or less uniform rate in a **pressure** producer while it operates and the gas is stored in tanks, generally of comparatively small capacity, from which it is drawn to meet the varying needs of the engine. A **suction** producer operating without an auxiliary "exhauster" is not provided with a storage tank, but the gas is generated at the rate demanded by the needs of the engine.

Producer Gas. The most common method for making producer gas to be used in engines is to admit both air and steam (or water vapor) simultaneously and continuously to the incandescent fuel bed. Another method is to burn the fuel for a time with air alone; that is, without any steam, till the fuel bed becomes highly incandescent, and then shut off the supply of air and pass steam or water vapor into the fuel till its temperature becomes so low that very little gas is formed and the air must be used again with the steam supply shut off. The producer continues in operation by alternating the admission of air and steam to the fuel bed. The former of these two methods is the one most generally used.

Suction Gas Producer. Anthracite coal is the most satisfactory fuel for suction gas producers, some preferring "chestnut" size, while others get the most satisfactory operation with the "pea" size if the coal supplied is clean. A producer or

generator for such fuel is illustrated in **Fig. 218.** It consists essentially of a vertical cylinder of cast-iron **P** lined with fire-brick, and having grates near the bottom which are easily cleaned and stoked through doors conveniently located near the base. A side outlet is sometimes provided for the removal of ashes from below the grate; but in many cases the ashes are removed through the stoking doors. It is necessary that the producer should be air-tight except for the regular openings provided for the admission of air and steam or water vapor. In the usual forms of suction producers the " suction " stroke of the engine **E** draws the necessary supply of air and steam or

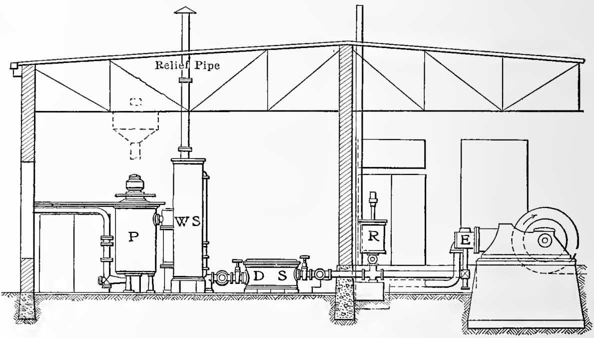


FIG. 218.—Suction Gas Producer and Engine.

vapor through the incandescent fuel bed, where the chemical changes of dissociating the steam and burning the coal occur. The fuel gas, as a rule, is conducted in a pipe from the top of the producer to the bottom of a " wet scrubber " **WS** for cleaning the gas by washing it with water. This scrubber is usually a vertical cylinder of cast-iron filled with broken coke, or else having a large number of wooden slats over which a small stream of water trickles from top to bottom. In its passage through the " wet " scrubber the soot, dust, tarry and other impurities are removed from the gas, after which it passes to a " dry " scrubber **DS,** known also as a " purifier " or " moisture separator," in which it passes through a mass of either excelsior, sawdust, or fine wood shavings, provided for removing moisture and

such solid particles as were not taken out in the " wet scrubber." After leaving the " dry scrubber " the gas should be properly cleaned for satisfactory use in the engine cylinder. A regulator or receiver **R** is needed for providing a volume of gas for expanding and flowing into the engine **E** during the suction stroke. By this means a comparatively steady flow of gas can be maintained from the producer and the resistance due to pipe friction to be overcome during the suction stroke is very much less than it would be if no reciever were used. Many producer plants are

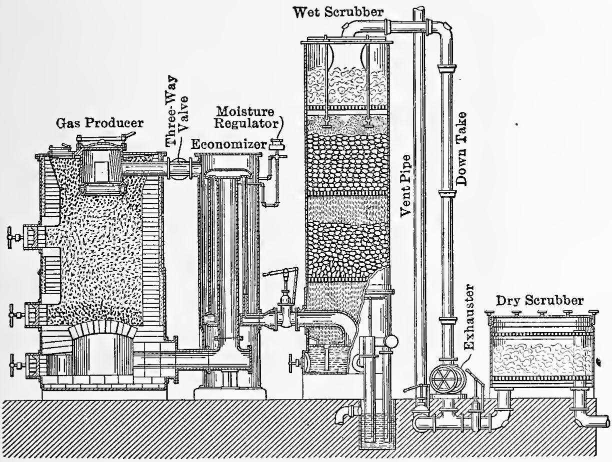


FIG 219.—Section of Suction Gas Producer with Economizer.

provided with an **economizer** located between the producer and the wet scrubber, as shown in **Fig. 219.** With this arrangement the air before going to the producer passes through this economizer, where it receives heat absorbed from the hot producer gas. The exhaust from the gas engine is also sometimes used for heating the air before it enters the producer. As a result of the chemical changes with an insufficient supply of air for complete combustion taking place in a producer of the type just described, some of the carbon in the coal is not burned completely; that is, to carbonic dioxide, CO_2, but forms instead

carbonic oxide, CO. The excess of carbon then combines with the oxygen liberated by the decomposition of steam into its elements (hydrogen and oxygen). The following reaction shows the formation of carbonic oxide from carbon and steam:

$$\begin{array}{ccccl} 2H_2O + & C_2 = & 2CO + & 2H_2 & \\ 2 & 1 & 2 & 2 & \text{volume proportions.} \\ 36 & 24 & 56 & 4 & \text{weight} \quad `` \end{array}$$

This reaction shows further that the volumes of carbonic oxide and hydrogen produced by the decomposition of steam are equal to each other, and that the total volume of these two gases, leaving out of account now the carbonic oxide formed by incomplete combustion, is twice as large as that of the steam used, considering, of course, equal temperatures and pressures.

In the decomposition of steam in the producer more heat is absorbed than is produced by the combustion of its carbon to carbonic oxide.[1] The extra heat required to make the process continuous, is provided by the combustion of part of the carbon of the fuel in air as described above. From this source the heat carried off in the gas leaving the producer, that lost by radiation, etc., is produced.

In the type of suction producer shown in **Fig. 218,** the steam is generated in a **vaporizer** which is simply a circular trough of cast iron encircling the top of the producer on the inside. This arrangement permits a constant gas supply automatically regulated to the needs of the engine. When the temperatures of the fire and of the gas passing from it are raised more heat will be conducted to the vaporizer and the rate of steaming will be increased; but this increased amount of steam delivered into the fuel bed will cool the fire to the proper temperature. In this way the temperature and composition of the gas can be kept fairly constant if the producer is otherwise operating properly. In **Fig. 219** the vaporizer is in the economizer, but here also the rate of steaming is regulated by the temperature of the gas leaving the producer, so that the regulation is also automatic.

[1] When a pound of carbon is burned to carbonic oxide (CO) only 4400 B.T.U. are produced, while when the same weight of carbon is burned to carbonic dioxide (CO_2) the heat developed is 14,650 B.T.U.

In some producers the steam is admitted through pipes entering the fuel space some distance above the grate bars. This arrangement is used to secure more perfect combustion of the fuel than is possible in the usual arrangement, because now only air is permitted to come into contact with the fuel in the lowest and hottest part of the fuel bed, and a high temperature can be maintained without difficulty. In this way it is possible to improve the economy of producers which otherwise would be operated, so that there would be a considerable amount of unburned coal removed with the ashes.

An air blower or enclosed fan of some kind is required to create a draught for starting the fire or for bringing up the fire after it has been "idle" for some time. When blowing up the fire a vent is opened between the producer and the wet scrubber to allow the gas generated to pass off, preferably through a chimney. If the plant is provided with an economizer the vent should be located so that the economizer will be heated during the time required for blowing up the fire.

Capacity and Efficiency of Gas Producers. The important result to be determined from tests of a gas producer is the ratio of the **heat value of the gas produced** (in B. T. U.) to the **heat value in the same units of the fuel used** and the mechanical or electrical energy required in producing the gas. The capacity or the rate at which gas can be produced is also important, since a high rate of gasification means lower initial costs of the plant. In reports of tests of gas producers it should be clearly stated whether the high or the low heat value of the gas has been used in the calculations. There is no accepted rule as to whether the high or the low heat value should be used in guarantees, so that the one to be used must be clearly stated. Probably the best method of stating guarantees is to give the amount (volume) of gas at a standard temperature and pressure and the heat value (high or low as perferred) per unit volume (usually a cubic foot) that a producer and its accessories will deliver from a stated weight of coal, of which the heat value per pound is also given. Mechanical, electrical, or other energy received from outside sources must also be taken into account. In the specifications for guarantees it should be stated that the loss of unburned fuel in the ash is to be charged as fuel used by the producer.

Commercial and Grate Efficiency. The efficiencies of a producer gas plant may be stated in a number of ways, but most of the items included will vary with the kind of producer used, so that what is known as the **commercial efficiency** $\mathbf{E}_c$ is the only important one for comparison. Using then the following symbols,

$\mathbf{h}_g$ = heat value of gas made B.T.U. per hour;

$\mathbf{h}_f$ = heat value of fuel used B.T.U. per hour;

$\mathbf{h}_o$ = heat equivalent of energy from outside sources, B.T.U. per hour;

$\mathbf{h}_b$ = heat of fuel actually burned in producer, B.T.U. per hour.

$$\mathbf{E}_c = \frac{\mathbf{h}_g}{\mathbf{h}_f + \mathbf{h}_0}, \qquad (90)$$

and the **efficiency** of the grates $\mathbf{E}_g$ may be stated as

$$\mathbf{E}_g = \frac{\mathbf{h}_b}{\mathbf{h}_f} \qquad (91)$$

The following data of observations and results should be included in a report of a test.

1. Duration of test, hours..
2. Brake horse power..
3. Coal fired per hour, lbs..
4. Coal fired per brake horse power per hour, lbs....................
5. Coal fired per square foot [1] per hour, lbs........................
6. Cooling water per hour, cu.ft..
7. Cooling water per brake horse power per hour, cu.ft...............
8. Inlet temperature of cooling water, deg. F........................
9. Outlet temperature of cooling water, deg. F........................
10. Average temperature rise, deg. F..
11. Heat value of gas by..........calorimeter,[2] B.T.U...............
12. Heat value of gas, maximum, B.T.U........................
13. Heat value of gas, minimum, B.T.U........................
14. Suction at scrubber, ins. of water........................
15. Temperature of gas leaving producer, deg. F....................
16. Temperature of gas leaving economizer, deg. F...................
17. Mechanical efficiency of engine, per cent........................
18. Thermal efficiency of engine, per cent........................
19. Efficiency of producer, based on lower heat value of gas, per cent...
20. Heat balance..

[1] Rate of gasification per sq. ft. of fuel bed area.

[2] Calorific values are all reduced to standard conditions used for **commercial** tests at 62 deg. F. and 30 inches of mercury (barometer).

CHAPTER XV.

TESTING OF VENTILATING FANS OR BLOWERS AND AIR COMPRESSORS

Ventilating Fans or Blowers are classified in general into one of the three groups designated as follows:

(1) Centrifugal fans;
(2) Disk (Propeller) fans;
(3) Turbine (Sirocco) fans;
(4) Positive pressure blowers.

Centrifugal Fans are used almost exclusively when large volumes of air are to be handled at a comparatively small pressure. Such a fan consists essentially of a number of plates,

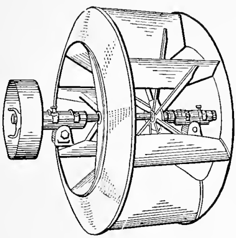


Fig. 222.

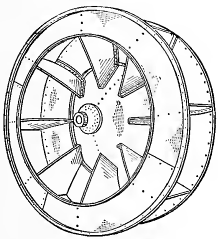


Fig. 223.

" Standard " Types of Ventilating Fans.

either flat or curved, attached to radial arms springing from a central hub through which the driving shaft passes, as in the " spider " type shown in **Fig. 222,** or the blades may be attached to a conical plate as in **Fig. 223.** Fans resembling either of these two designs are known commercially as the **" standard "** type. The " width " of the blades is, in most cases, parallel to the shaft.

The work performed by a centrifugal type of fan is equal to the resistance times the velocity of flow. Since, however, the fan resistances are proportional to the square of the velocity,[1] the work done is proportional to the cube of the velocity. Fans of this type are invariably provided with an air-tight spiral casing **C** as shown in **Fig. 224,** where the suction is at **S** and the discharge is at **D.** The fan shown in this figure represents the celebrated Rateau designs, which are extensively used in Europe.

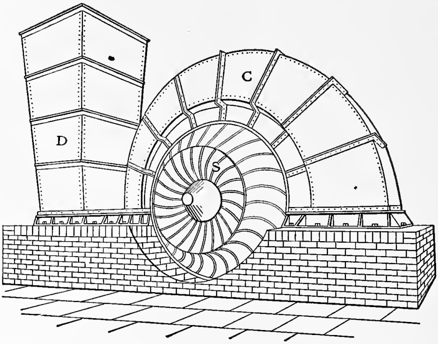


Fig. 224.—Rateau Ventilating Fan Showing Spiral Casing.

Disk or Propeller Fans are best illustrated by the so-called "electric" fans so commonly used in offices, shops and dwellings. Fans of this type are usually of a very light construction with the vanes arranged as in a screw propeller for a ship. In many

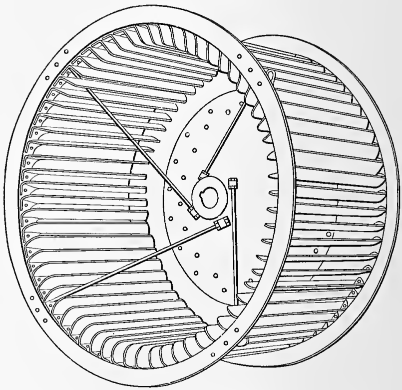


Fig. 225.—Turbine Type (Sirocco) Fan.

[1] See Professor Rateau's articles in *Revue de Mécanique*, vol. 1, pages 629–837.

cases fans of this type are not provided with casings, so that it is more difficult to make velocity measurements than with centrifugal fans.

Turbine or " Sirocco " fans have an **impeller** or fan wheel of the " squirrel cage " type, as illustrated in **Fig. 225.** Fans of this type can be designed to give very high efficiencies. This is due primarily to two characteristic features adopted in these designs. By the use of very short blades a very large intake space for the suction is provided which is practically unobstructed, thus giving a very free " suction." The other important feature of this fan is that the air leaves the blades

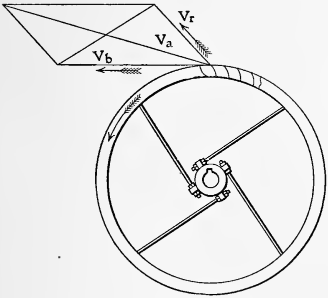


FIG. 226.—Velocity Diagram for a Turbine Fan.

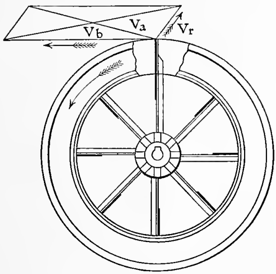


FIG. 227.—Velocity Diagram for a " Standard " Fan.

at a higher velocity than that at which the tips of the blades are moving. The importance of this result is shown by a comparison of **Figs. 226** and **227.** The former illustrates the type of blading in a turbine or "Sirocco" fan and shows graphically by a velocity diagram, constructed like a parallelogram of forces, the velocity of the tips of the blades $\mathbf{V}_b$, the velocity of radial flow **in** the blades $\mathbf{V}_r$ and the **absolute** velocity of the discharge $\mathbf{V}_a$, which is the velocity of the air with respect to the stationary casing. It will be observed that in **Fig. 226,** $\mathbf{V}_a$ is nearly 50 per cent greater than the velocity of the tips $\mathbf{V}_b$, while in **Fig. 227,** representing the corresponding velocities for a standard type of fan, the absolute velocity of the discharge $\mathbf{V}_a$ is actually considerably less than the speed of the tips of the blades. Increased velocity is accomplished in a type of

fan like **Fig. 226,** not only by the curvature of the tips of the blades, but also to some extent by making the blades somewhat concave with the inner ends (toward the center) practically radial. By this method of designing the outer edges of the blades have a smaller space between them than the inner edges. This has the effect of reducing the area on the discharge side of the blades and consequently the velocity of the air is increased.

Positive Pressure Blowers are used principally for blast furnaces and smelters where a higher pressure is needed than can be efficiently obtained with a centrifugal fan. A section showing the rotors and casing of one of the blowers is shown in **Fig. 228.** It is often called Root's blower. The efficiency of a blower of this type depends on the accuracy of the fitting of the two rotors, **A** and **B,** both with respect to each other and to the casing. It is for this reason that when new the efficiency is high, but after being in service for several years the bearings and the surfaces of the rotors will become worn, so that there is considerable leakage and consequent loss of efficiency.

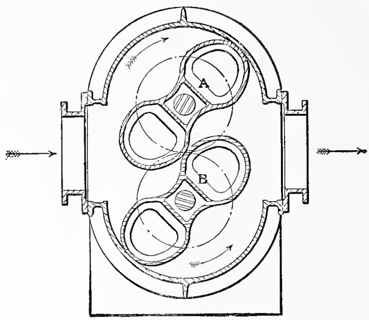


Fig. 228.—Typical Positive Pressure Blower.

Tests of Ventilating Fans or Blowers are made usually by a very simple method; that is, by determining the necessary data for calculating efficiency by measuring the work done by the fan "on the air" in giving velocity, and the power required to drive the fan alone, excluding bearing friction. The fan is preferably operated by an electric motor, of which the efficiency can be readily determined by a Prony brake test. The power required to overcome the bearing friction of the shaft of the fan may well be first determined by measuring the power input [1]

[1] For normal operation "friction work" is, for machinery in general, proportional to the speed.

(kilowatts) for a series of speeds when the keys fastening the fan to its shaft have been removed and the fan itself has been "blocked" in its casing. The shaft will then revolve in the hub of the fan and in the bearings. After attaching the fan again to the shaft the input to the motor and the work done by the fan "on the air" should be determined for various speeds. Then obviously the ratio of the work done by the fan divided by the power required to drive it after correction for the efficiency [1] of the motor and bearing friction is the actual efficiency of the fan. In general terms this may be stated as follows:

f = input to motor to drive motor and shaft of fan in bearings, kilowatts;

i = input to motor to drive motor and fan, in kilowatts;

e = efficiency of motor for motor input of **i** kilowatts and at speed of test;

e′ = efficiency of motor for motor input of **f** kilowatts and at speed of test.

Then if $\mathbf{i}_n$ is the net work in **horse power** to drive fan alone,

$$\mathbf{i}_n = \frac{\mathbf{ei} - \mathbf{e'f}}{0.746}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (92)$$

If the fan to be tested is direct-connected to a steam engine, the test is usually made by measuring the indicated horse power of the engine at the various speeds and also with the fan disconnected for no load. If it is possible to do so the fan should be removed from the shaft for the no-load test to determine bearing friction.

The work done by the fan "on the air," is most readily calculated by the same method used to calculate the efficiency of hydraulic pumps (see page 357), that is, by multiplying the number of pounds of air delivered per unit of time by the head in feet of air corresponding to the discharge pressure. This product is obviously in terms of work in foot-pounds per unit of time, and dividing by 33,000 the corresponding horse power is obtained. Using the following symbols, the same results may be expressed, however, by the product of "pressure times volume" as follows:

[1] Motor efficiency must be necessarily determined for the conditions of each test; that is, for the same kilowatts and speed as for each test.

v = velocity of air in feet per second;
h = head in feet of air necessary to produce a velocity of **v** feet per second, or,
= water pressure **p** in **inches** observed with a manometer, produced by the **velocity** of the air times the ratio.

$$\frac{wt.\text{ of a cubic foot of water}}{wt.\text{ of a cubic foot of air}^{1}}$$

$$\mathbf{v} = \sqrt{2gh} = \sqrt{2g\frac{\mathbf{p}\times 62.3}{\text{wt. cu. ft. air for test}}} \quad . \quad . \quad . \quad (93)$$

and $\mathbf{V}_m$ = velocity in feet per **minute** is (taking $2g = 64.3$)

$$\mathbf{V}_m = 1096.4\sqrt{\frac{\mathbf{p}}{\text{wt. cu. ft. air for test}}} \quad . \quad . \quad . \quad (94)$$

Now the velocity in feet per minute $\mathbf{V}_m$ multiplied by the area of the section at which the velocity was observed, gives cubic feet **C** of air discharged per minute, and if **P** is the total pressure (static + velocity) in pounds per square **foot** then we have for **j** the "**air horse power**" or the work done by the fan "on the air."

$$\mathbf{j} = \frac{\mathbf{CP}}{33{,}000}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (95)$$

and efficiency of fan **E**, is

$$\mathbf{E} = \frac{\mathbf{j}}{\mathbf{i}_n} = \frac{\mathbf{CP}}{33{,}000\ \mathbf{i}_n}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (96)$$

Velocity measurements are usually made with a **Pitot tube** consisting essentially as shown in Figs. 136 to 138, pages 143–144, of two tubes with openings at the end, arranged so that one of them "faces" in the direction of flow and the other extends in a radial direction. The former is subjected to the sum of the velocity and static pressures, while the latter receives only the static pressure.

[1] Weight of air taken for calculation must be that corresponding to the **total** pressure in the discharge pipe, the temperature and the humidity. For tables of weight of air see pages 145 and 331, also Kent's "Mechanical Engineers' Pocket-Book," 8th edition, pages 583–588, and "Calculating and Testing Ventilating Systems," issued by U. S. Navy Department, Washington.

The following table of relative humidity for determinations with a wet- and a dry-bulb thermometer is used by the U. S. Weather Bureau:

TABLE OF RELATIVE HUMIDITY, PER CENT

Dry Thermometer, Deg. F.	Difference between the Dry and Wet Thermometers, Deg. F.																										
	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	26	28	30
	Relative Humidity, Saturation being 100. (Barometer = 30 ins.)																										
32	89	79	69	59	49	39	30	20	11	2																	
40	92	83	95	68	60	52	45	37	29	23	15	7	0														
50	93	87	80	74	67	61	55	49	43	38	32	27	21	16	11	5	0										
60	94	89	83	78	73	68	63	58	53	48	43	39	34	30	26	21	17	13	9	5	1						
70	95	90	86	81	77	72	68	64	59	55	51	48	44	40	36	33	29	25	22	19	15	12	9	6			
80	96	91	87	83	79	75	72	68	64	61	57	54	50	47	44	41	38	35	32	29	26	23	20	18	12	7	
90	96	92	89	85	81	78	74	71	68	65	61	58	55	52	49	47	44	41	39	36	34	31	29	26	22	17	13
100	96	93	89	86	83	80	77	73	70	68	65	62	59	56	54	51	49	46	44	41	39	37	35	33	28	24	21
110	97	93	90	87	84	81	78	75	73	70	67	65	62	60	57	55	52	50	48	46	44	42	40	38	34	30	26
120	97	94	91	88	85	82	80	77	74	72	69	67	65	62	60	58	55	53	51	49	47	45	43	41	38	34	31
140	97	95	92	89	87	84	82	79	77	75	73	70	68	66	64	62	60	58	56	54	53	51	49	47	44	41	38

By using the above table the weight of a cubic foot of air for any degree of saturation and temperature can be easily calculated from the tables of the weight of **dry** air and **100 per cent saturated** air [1] as given on page 145.

Anemometers are also frequently used for velocity measurements of air, but they are not generally so reliable as good Pitot tubes. Since, however, the observations can be taken directly in feet per minute these instruments are used for nearly all work where no great accuracy is expected.

An example showing the method of calculation for **j**, the work done by the fan "on the air," may assist in making the method of calculation clearer.

A series of Pitot-tube measurements taken at ten different places in the cross-section of an air duct shows that the "velocity" pressure was .795 and the total pressure 1.09 inches of water. The observations of barometric pressure and temperatures by wet- and dry-bulb thermometers, together with the "total" pressure given above, served to determine the density or the weight of a cubic foot of air at the conditions of the test. Barometric pressure

[1] Somewhat more accurate determinations of the weight of air can be calculated by empirical formulas given in Kent's "Mechanical Engineers' Pocket-Book," 8th edition, pages 583–588.

was 29.40 inches of **mercury** and temperatures of wet- and dry-bulb thermometers were respectively 54 and 71 degrees Fahrenheit. According to the tables of the properties of air (see footnote, page 330) this was .07329 pound per cubic foot. Velocity of the air $\mathbf{V}_m$ in feet per minute is therefore,

$$\mathbf{V}_m = 1096.4\sqrt{\frac{.795}{.07329}} = 3625 \textbf{ feet per minute.}$$

The diameter of the pipe was 10 inches, of which the area is 0.545 square foot. Cubic feet air discharged per minute (**C**) are 3625×0.545 or 1978. The total pressure **P** is the "total" pressure[1] in pounds per square foot or $\left(\frac{1.09}{13.6}\right) .491 \times 144$ or 5.66. Work done by the fan "on the air" is, then,

$$\mathbf{j} = \frac{1978 \times 5.66}{33{,}000} = 0.34 \textbf{ horse power.}$$

Efficiency tests should be made with the fan operating under the discharge pressure for which it was designed or for which the guarantee was made as the case may be. The efficiency of the fan may be considerably higher with a lower discharge pressure than when connected up in a ventilating system where the discharge pressure is comparatively high.

Testing Ventilating Systems. When tests are to be made of ventilating systems precautions should be taken in the examination of all ducts and piping to see that they are clear of all lumber, rubbish, etc., and that the dampers are properly set. The tests consist usually in measuring in each system the "static" and the "total" pressures with a Pitot tube with all louvers open. These tests should be made with the fan running at high, low and three or four intermediate speeds. All the results should be checked by plotting a curve with revolutions per minute for abscissas and cubic feet of air delivered per minute for ordinates. This curve should be approximately a straight line

[1] In this expression 13.6 is the specific gravity of mercury, .491 is a factor for changing inches of mercury at room temperature to pounds per square inch, and 144 is used to change pounds per square inch to pounds per square foot.

passing through the origin if all the louvers have remained open throughout the tests. On the same abscissas curves of pressure for ordinates will also be of advantage in showing the consistency of observations. The location selected for the testing slot to be used for inserting the Pitot tube in the mains should be as near as possible to the fan; preferably no branches, however small, should run off between the fan and the testing slots in the mains. Furthermore the testing slots should not be near turns and bends and particularly no turns or elbows should be immediately ahead of a slot, that is, in the direction toward the fan. These testings lots should be covered when not in use.

In the **U. S. Navy Department the standard conditions** adopted for testing ventilating fans and air-supply mains are a pressure of 5 pounds per **square foot** in the moving air at the discharge outlet from the fan and a velocity of 2000 feet per minute. Air at the standard conditions is to be at 70 degrees Fahrenheit and a relative humidity of 70 per cent. Under these standard conditions a cubic foot of air weighs .07465 pound. The pressure of 5 pounds is equivalent to a **pressure** head of 67 feet of standard density air. A velocity of 2000 feet per minute corresponds to a **velocity** head of 17.27 feet. Total head against which the air is delivered to the supply mains for the standard conditions is then 84.27 feet, making a very satisfactory combination of velocity and pressure head, approaching as it does the maximum possible delivery for this pressure head.

Corrections for Losses of Total Head in Ducts. There is always some loss of total head along a duct or pipe due to friction. As a result there is a smaller delivery than that given for standard conditions. Using the following symbols:

h_f = loss of head in feet due to friction;

f = coefficient of friction = .0008 in piping of good construction;

l = length of duct in feet;

d = diameter of duct in feet;

V_m = velocity of flow through duct in feet per **minute.**

$$h_f = \frac{V_m^2 l}{11{,}250{,}000 d} \quad \ldots \ldots \quad (97)$$

If $V_m = 2000$ then $h_f = .3556\frac{l}{d}$.

Loss of head in a **square duct** is usually assumed to be the same as for a round one; but for a duct of **rectangular section** of which the short side is **b** and the long side is **nb,** the formula above becomes, using **l** and $\mathbf{V}_m$ as before,

$$h_f = \frac{1+n}{n} \times \frac{l}{b} \times \frac{V_m^2}{2,250,000}. \quad . \quad . \quad . \quad . \quad (98)$$

With the help of these formulas when the size of the main ducts and the discharge in cubic feet per minute at each outlet are known, the head at each outlet as compared with the standard total head of 84.27 feet can be calculated. As a "rough and ready" rule it is often stated that for a loss of one foot of head there is a loss of six-tenths per cent delivery (cubic feet per minute).

Testing Air Compressors. The various types of machines for compressing air are usually operated either by a steam engine or by an electric motor. Power delivered to the compressors by the engine or motor is therefore measured by one of the methods outlined above for ventilating fans. Work done "on the air" may be measured by a Pitot tube or by an anemometer in a duct if the discharge pipe is large enough as it leaves the compressor.

Air compressors, particularly of the reciprocating type, are designed, as a rule, for operation at considerably higher pressures than would be suitable for ventilating fans, and the volume delivered must usually be measured in a comparatively small pipe. Air at high pressure is generally measured by calculating the flow through an orifice, carefully rounded on the "entrance" side, by **Fliegner's formula:**

$$w = .530 \frac{ap_1}{\sqrt{T_1}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (99)$$

where **w** =weight of air discharged per **second** in pounds;
a =least area of the orifice in square inches;
$\mathbf{p}_1$ =the initial pressure on the orifice (before expansion in the orifice) in pounds per square inch absolute.
$\mathbf{T}_1$ =absolute temperature corresponding to the pressure $\mathbf{p}_1$.

This formula has been found to be very accurate for the conditions of average practice, but is **not to be used when p_1 is less than twice the atmospheric pressure.**

The weight of flow, **w** pounds per second multiplied by the head of air in feet corresponding to the " total " pressure, gives the foot-pounds of work " done on the air " per second; and this product divided by the **net power** required to drive the compressor is the **mechanical efficiency.**

In an air compressor in which the air cylinder is direct connected to the steam cylinder the net power required to drive the compressor is determined by finding the indicated horse

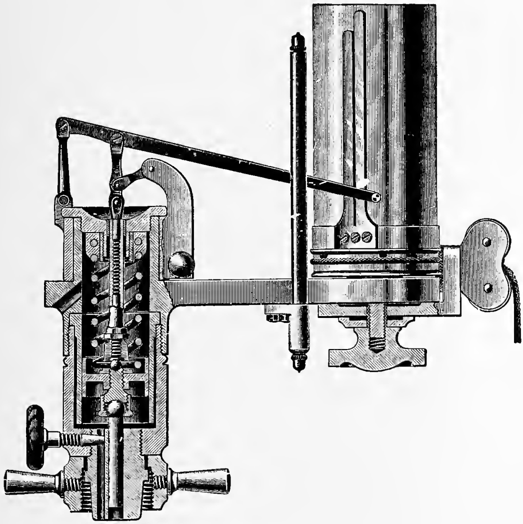


FIG. 230.—Crosby High-pressure Indicator (Ordnance Type).

power of the air cylinder and adding the friction in the air cylinder, which in many cases can be assumed to be half the difference between the indicated horse power as measured in the steam and air cylinders. For use on air compressors operating at high pressures special indicators are made. One of the best is made by the Crosby Steam Gage and Valve Co., Boston, Mass., and is illustrated in **Fig. 230.** It is similar in design to the gas engine indicator illustrated in Fig. 211, page 308, except that the piston in the lower cylinder of the indicator is very

small, only one-fortieth of a square inch in area. This indicator can therefore be readily used for pressures as high as 10,000 pounds per square inch.

Another method for measuring the volume of the delivery of the compressor is to connect the discharge pipe to a large air-tight tank, provided, however, with an opening near the bottom through which water can be removed. If this tank has been filled with water before the test begins and the pressure is maintained constant in it by taking out water as air is being pumped in, then the cubic feet of water taken out is the same as the volume of air put in. This method is also serviceable for calibrating nozzles to test the accuracy of Fliegner's formula. It will be necessary to use a gage graduated as accurately as possible, and if the pressure is not very high a mercury column will be most satisfactory, as it will show more quickly small variations in pressure.

Test of an Air Compressor

1. Revolutions per minute. . . . .
2. Mean effective pressure, steam cylinder, lbs. per sq.in. . . . .
3. Mean effective pressure, air cylinder, " " " . . . .
4. Indicated horse power, steam cylinder. . . . .
5. Indicated horse power, air cylinder. . . . .
6. Mechanical efficiency, per cent. . . . .
7. Steam pressure, lbs. per sq.in. . . . .
8. Reservoir pressure, " . . . .
9. Nozzle pressure, " . . . .
10. Temperature of air entering cylinder, deg. F. . . . .
11. Temperature of air leaving cylinder, deg. F. . . . .
12. Temperature of air at nozzle, deg. F. . . . .
13. Temperature of air "outside," deg. F. . . . .
14. Temperature of steam calorimeter, deg. F. . . . .
15. Temperature of water entering jacket, deg. F. . . . .
16. Temperature of water leaving jacket, deg. F. . . . .
17. Weight of jacket water, lbs. . . . .
18. Weight of condensed steam, lbs. . . . .
19. Heat absorbed by jacket water, B.T.U. . . . .
20. Quality of steam. . . . .
21. Steam used per I.H.P. per hour, lbs. . . . .
22. Theoretical air discharged per piston displacement, cu.ft. per hour (at 32 deg. F. and 14.7 lbs. per sq.in.). . . . .
23. Actual air discharged through nozzle, cu.ft. per hour (at 32 deg. F. and 14.7 lbs. per sq.in.). . . . .
24. Slip, per cent $\frac{(22)-(23)}{(22)}$ . . . .
25. Volumetric efficiency $(23) \div (22)$. . . . .
26. Air discharged per I.H.P. per hour, lbs.

CHAPTER XVI

TESTING OF REFRIGERATION PLANTS

Refrigerating machines present a most interesting example of the conversion of heat energy. In the simplest forms these machines consist of a compressor driven by a steam engine, or other motive power, serving to compress a gas or vapor as the case may be. This gas or vapor is then passed under pressure through a surface condenser, where the cooling water absorbs the heat generated in the work of compression and then passes into an **expanding vessel** into which it discharges at a very low temperature. Now in order to vaporize any liquid, it is necessary to maintain a continual application of heat in order to bring about this physical change. To convert a unit weight of liquid to a unit weight of vapor at the same pressure the heat required is always a constant quantity for the same liquid. Thus, as a familiar example, to convert a pound of water at "atmospheric" pressure and 212 degrees Fahrenheit into steam at the same pressure and temperature requires the application of 970 B.T.U.; and conversely, to condense a pound of steam at this same pressure and temperature, it is necessary to abstract 970 B.T.U. by contact with a cold body. Steam as the working medium in a refrigerating machine would, of course, be impracticable, because the lowest temperature resulting from **actual condensation** in a workable plant would be very much above the freezing point of water; but there are a number of liquids which have a very much lower boiling point than water. Of these ammonia (NH_3), carbonic dioxide (CO_2), and sulphurous dioxide (SO_2) are successfully used for purposes of refrigeration. The use of all these depends on the absorption of their latent heat in their conversion from a vapor or gas to the liquid condition. In practice, the refrigerating medium most commonly used is anhydrous ammonia, although carbonic dioxide is also frequently

employed. The latter is preferred usually where ammonia gas might be dangerous or otherwise objectionable.

In the simplest form of refrigerating plant the necessary machinery consists of (1) a **compressor** to raise the gas to the necessary pressure; (2) a **surface condenser** to absorb by means of cooling water the heat generated by the mechanical work of compression; and (3) an **expanding or evaporating vessel** where the liquid is re-evaporated into a gas and, of course, absorbs heat in the operation. A very simple refrigerating machine is shown in **Fig. 231.** It consists of the compressor **C** discharging gas under pressure [1] through the pipe **P** into the

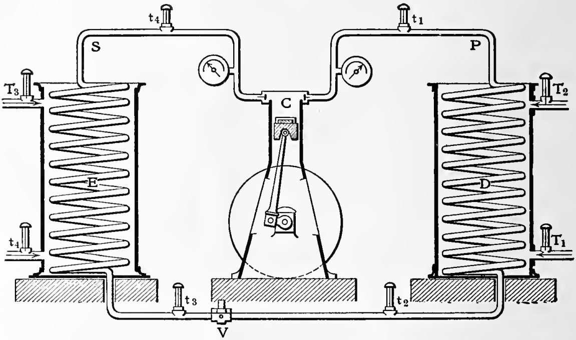


FIG. 231.—Typical Refrigerating Apparatus.

condensing coil **D,** consisting in this simple apparatus of a coil of pipe in a tank through which the cooling water circulates. An expanding valve **V** serves for reducing the pressure and evaporating the liquid coming from **D.** The expanding vessel or **evaporator E** consists of a coil of pipe immersed in a tank containing the liquid to be cooled. Drops of liquid accumulate in the bottom coils of the condenser **D,** to be discharged through the expanding valve **V** into the evaporator **E.** Since the compressor receives its supply of gas from the evaporator, the pressure in the latter must be less than in the condenser. On this

[1] In order to liquefy any gas or vapor, obviously it is necessary to bring the molecules closer together, and this can be accomplished either by increasing the pressure or decreasing the temperature or by both.

account, then, the liquid after expanding will begin to boil and will absorb heat from the surrounding liquid in its transformation into a gas. In such a process the temperature of the cooling liquid may become very low. The refrigerating liquid in the evaporator will be entirely gasified or vaporized and returns finally to the compressor **C** in this state through the suction pipe **S,** thus completing the cycle of operations.

After this brief explanation of the principles of the operation of a refrigerating machine we can take up a brief discussion of the thermal processes as regards the interchangeability of heat and work. Using the symbol **r** for the latent heat of vaporization of the refrigerating medium in B.T.U. per pound, **h** for the heat imparted by compression in the same units,[1] **w** for the weight in pounds of the gas or vapor entering the compressor in a given time, then, neglecting external losses, **wr** will represent the heat abstracted in the evaporator and $\mathbf{h}+\mathbf{wr}$ is the heat given to the cooling liquid in the condenser.

In the practical operation of a refrigerating plant the evaporation is maintained at a very low temperature, and some heat must necessarily be given to it by the refrigerating medium itself, since it enters the evaporator, in comparison, in a moderately warm condition. Now if the difference in temperature between the condenser and the evaporator is **t** degrees Fahrenheit, a pound of refrigerating medium will give to the evaporator **st** B.T.U., if **s** is the specific heat of the refrigerating medium; and further if **w'** is the weight of this medium in pounds passing into the evaporator in a given time then the heat abstracted from the evaporator by the cooling liquid is

$$\mathbf{wr} - \mathbf{x} - \mathbf{w'st},$$

where **x** is the heat in B.T.U. lost by **radiation.** The term **w'st** is comparatively small in practical machines. If there is no leakage then, of course, **w'** will be the same as **w.**

Anhydrous ammonia is most commonly used as the refrigerating medium. It is preferable to many other fluids because

[1] The ratio $\frac{\mathbf{r}}{\mathbf{h}}$ is often called the **coefficient of efficiency** of the refrigerating medium.

of its comparatively high latent heat [1] and low pressure of vaporization.

Carbonic dioxide (CO_2), commercially known as carbonic acid, is a colorless gas without odor when pure, and is furthermore quite innocuous and has practically no injurious effect on animal tissues. It is injurious only when the proportion of it in air becomes so large that there remains an insufficient amount of oxygen. On this account, therefore, it is much safer and suitable as a refrigerating medium than ammonia. This gas can be readily liquefied either by lowering its temperature or by increasing the pressure. At ordinarily low temperatures it can only remain in the liquid state when under considerable pressure. When the pressure is removed, the heat absorbed from surrounding bodies assists in the rapid evaporation of the liquid and these bodies become correspondingly colder by this loss of heat.

Carbon dioxide is used only to a limited extent, but it is found particularly desirable on shipboard because of the compactness of the compressor that it requires and its inoffensive character when a leak occurs.

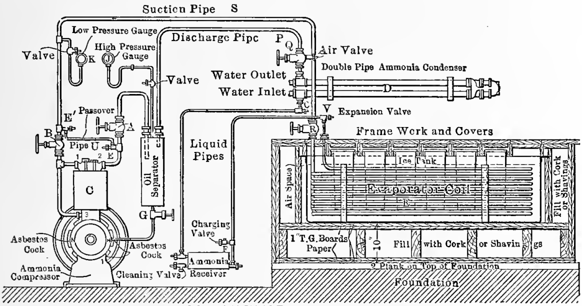
A typical **commercial refrigerating plant** for making ice and operating with a horizontal ammonia compressor is shown in **Fig. 232.** The same descriptive letters used in **Fig. 231** serve again for marking the important parts.

The efficiency of a refrigerating machine depends upon the difference between the extremes of temperature, but unlike heat engines, it has the greatest efficiency when the range of temperatures is small and when the final temperature is high.

When a change of volume of a saturated vapor is made under constant pressure in the presence of an excess of the liquid, the temperature remains constant. In this case the addition or absorption of heat to produce the change of volume causes an increase or decrease in the amount of the liquid mixed with the vapor. Vapors, even when saturated, if no longer in contact with their liquids, having heat added either by compression, by

[1] The latent heat of vaporization of ammonia is 555 B.T.U. at a temperature of zero degrees Fahrenheit, while that of carbon dioxide is only 123. The corresponding absolute pressures at the same temperature are 30 pounds per square inch for ammonia and 310 for carbon dioxide.

mechanical force or from an external source of heat, will behave practically like permanent gases and will become **superheated.** On this account refrigerating machines using liquefiable gas will give results differing according to the conditions of operation, depending primarily upon the state of the gas; that is, whether it remains constantly saturated or is superheated during a part of the cycle. Some ammonia plants are operated with an excess of liquid present during compression so that superheating is



COMPLETE ICE MAKING PLANT

FIG. 232. Refrigerating Plant with Ammonia Compressor.

prevented. This is known in practice as the "wet" or "cold" system of compression.

DENSITY OF LIQUID AMMONIA.[1]

At temp. deg. C..	−10	−5	0	5	10	15	20
At temp. deg. F..	14	23	32	41	50	59	68
Density........	0.6492	.6429	.6364	.6298	.6230	.6160	.6086

LATENT HEAT OF EVAPORATION OF AMMONIA

$$h_e = 555.5 - .613T - 0.000219T^2 \text{ (in B.T.U. degrees F.)}$$

Ledoux found $h_e = 583.33 - .5499T - 0.0001173T^2$ (in B.T.U. degrees F.)

For experimental values at different temperatures determined by Professor Denton, see *Transactions American Society Mechanical Engineers*, vol. 12, page 356. For calculated values, see vol. 10, page 646.

[1] These results may be expressed very nearly by

$= 0.6364 - 0.0014t$ degrees Centrigrade.

$0.6502 - 0.000777t$ degrees Fahrenheit.

Specific Heat and Available Latent Heat of Hot Ammonia

Latent heat at 15.67 lbs. and 0 degrees F. = 550.5 B.T.U. Specific heat = 1.096 − 0.0012 T (degrees).

Values at 15.67 lbs. Gage Pressure (Lucke)

Temperature of Liquid Supply. Deg. F.	Specific Heat.	Correction for Cooling. B.T.U.	Available Latent Heat for Saturated Vapor B.T.U. per lb.
5	1.090	5.45	550.05
10	1.084	10.84	544.66
15	1.078	16.17	539.33
20	1.072	21.44	534.06
25	1.066	26.65	528.85
30	1.060	31.80	523.70
35	1.054	36.89	518.61
40	1.048	41.92	513.68
45	1.042	46.89	508.61
50	1.036	51.80	503.70
55	1.030	56.65	498.85
60	1.024	61.44	494.06
65	1.018	66.17	489.33
70	1.012	70.84	484.66
75	1.006	75.45	480.05
80	1.000	80.00	475.50
85	.994	84.49	471.01
90	.988	88.92	466.58
95	.982	93.29	462.21
100	.976	97.60	457.90

The latent heat of saturated ammonia vapor as given by Lucke must be corrected in three ways: (1) For the temperature of the liquid, which must be cooled from its initial temperature to the temperature corresponding to the suction or back-pressure; (2) for wetness of the vapor, for which the correction is 5.555 B.T.U. for each per cent of moisture; (3) for superheat of vapor in case it leaves the expansion coil (evaporator) at a higher temperature than that corresponding to the pressure. This correction is additive and is approximately the number of degrees of superheat times the specific heat of superheated ammonia gas taken as 0.508.

Leakages of ammonia gas are very objectionable and may be dangerous. One of the most convenient and reliable means for locating a small leak is to burn a little sulphur at the end

of a stick. Where the sulphur fumes come into contact with the ammonia gas a white vapor is observed.

Efficiency of Refrigerating Machines. The maximum theoretical efficiency $\mathbf{E}_m$ of a refrigerating machine is expressed by the ratio,

$$\mathbf{E}_m=\frac{\mathbf{T}_0}{\mathbf{T}_1-\mathbf{T}_0} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (100)$$

where $\mathbf{T}_1$ is the highest and $\mathbf{T}_0$ is the lowest absolute temperature of the refrigerating medium. Another and more practical way to express the efficiency of a refrigerating plant is found by using as a basis the amount of fuel consumed and the "ice-melting" capacity[1] of the plant. If we use the following symbols:

$\mathbf{R}$ =refrigeration or "ice-melting" capacity per pound of fuel, in **pounds;**
w_b =pounds of brine circulated per hour, pounds;
s_b =specific heat of brine;
t_1 =temperature of brine entering expansion coils, deg. F.;
t_2 =temperature of brine leaving expansion coils, deg. F.;
w_f =fuel used per hour, pounds;

$$\mathbf{R}=\frac{w_b s(t_2-t_1)}{144 w_f}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (101)$$

and the **capacity C** of a machine in tons, of 2000 pounds, of refrigeration or ice-melting per 24 hours is

$$\mathbf{C}=\frac{24 w_b s(t_2-t_1)}{144\times 2000} \quad . \quad . \quad . \quad . \quad . \quad . \qquad (102)$$

Volumetric Efficiency. The ratio of the actual volume of ammonia discharged from the compressor to that calculated from the piston displacement is called the volumetric efficiency. The following formula deduced from Voorhees[2] gives in most

[1] Ice-melting capacity is a term applied to represent the cold produced in an insulated bath of brine, measured by the latent heat of fusion of ice, which is 144 B.T.U. per pound. More accurately it is the heat required to melt a pound of ice at 32 degrees Fahrenheit to water at the same temperature. The capacity of a machine in pounds or tons of "ice-melting" or of "refrigeration" does not mean that the machine would make that amount of ice; but that the cold produced is equivalent to the melting of the weight of ice to water.

[2] "Ice and Refrigeration" (1902).

practical cases the volumetric efficiency $\mathbf{E}_v$ of an ammonia compressor with a remarkable degree of accuracy:

$$\mathbf{E}_v = \frac{1-(t_1-t_0)}{1330} \quad . \quad . \quad . \quad . \quad . \quad . \quad (103)$$

where t_1 is the **theoretical** temperature of the gas after compression, t_0 is the temperature of the gas delivered to the compressor. Here t_1 can be calculated from the general equation for adiabatic compression where

$$t_1 + 460 = (t_0 + 460)\left(\frac{p_1}{p_0}\right).024 \quad . \quad . \quad . \quad . \quad (104)$$

Here p_1 and p_0 are the absolute pressures of the gas corresponding respectively to the temperatures t_1 and t_0. The actual temperature of the gas discharged from the compressor will be usually considerably, sometimes 50–60 degrees Fahrenheit, less than the theoretical.

Lucke[1] has deduced the following formula for the **indicated horse power (I.H.P.) required per ton of refrigerating capacity,** expressed in the following symbols:

$\mathbf{p}$ = the mean effective pressure in lbs. per square inch;
$\mathbf{l}$ = the length of the stroke in feet;
$\mathbf{a}$ = the area of the piston in square inches;
$\mathbf{n}$ = the number of compressions per minute;
$\mathbf{E}_v$ = the volumetric efficiency, as defined above;
$\mathbf{w}_c$ = the weight of a cubic foot of ammonia vapor at the back pressure as it exists in the cylinder when compression begins; $\mathbf{v}_c$ is the latent heat of vaporization available for refrigeration (see table page 342); 288,000 = the B.T.U. equivalent to one ton of refrigeration; that is, 2000×144; $\mathbf{c}$ = refrigerating capacity in tons per twenty-four hours. Then,

$$\text{I.H.P.} = \frac{\dfrac{\mathbf{plan}}{33{,}000}}{\dfrac{\mathbf{laE}_v\mathbf{nw}_c \times \mathbf{v}_c \times 60 \times 24}{144 \times 288{,}000}} \quad . \quad . \quad . \quad (105)$$

$$= \frac{0.87}{\mathbf{w}_c\mathbf{v}_c} \times \frac{\mathbf{p}}{\mathbf{E}_v}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (106)$$

[1] *Proceedings American Society of Refrigerating Engineers* (1908).

Ammonia Absorption Refrigerating Machines. Another class of refrigerating apparatus, operating by what is known as the absorption system, has been installed in some places. It consists of a generator containing a concentrated solution of ammonia in water. This generator is heated usually by means of a coil of pipes taking live steam from a boiler, although frequently the exhaust steam from engines is utilized to advantage. In this system a weak ammonia vapor passes first into an "analyzer" where some of the water is separated from the ammonia vapor and then into a "rectifier," where the concentrated vapor is cooled, precipitating still more water, and then discharges into the condenser coils. The lower coils of the condenser are connected to the upper part of the "cooler" or brine tank. An absorption chamber is provided which is filled with a weak solution of ammonia, and this chamber is also connected with the cooling-tank. The absorption chamber communicates with generator by two tubes, one going to the bottom of the generator from the top of the chamber, and the other from the bottom of the chamber to the top of the generator. In the latter pipe line, a pump is located to force the liquid from the absorption chamber, where the pressure is about atmospheric, to the generator, where the pressure is from 100 to 200 pounds per square inch. In the operation of this apparatus the ammonia and water in the generator are first heated by the coil of steam pipes, and as the ammonia is freed from the solution the pressure rises. When this pressure attains that of saturated vapor at the temperature of the condenser it becomes liquefied, condensing also a small amount of steam. A suitable expansion valve regulates the flow of the liquefied gas into the refrigerating coils in the cooler. As it escapes into these coils it expands and is again vaporized, absorbing heat from the liquid or gas required to be cooled. Just as rapidly as vaporization goes on the gas is absorbed by the weak solution in the absorbing chamber. The heat in the generator has the effect of separating a strong from a weak solution, the greater concentration being in the upper part. The weaker portion of the solution is conveyed by the pipe entering the top of the absorption chamber. The satisfactory operation of this apparatus depends upon careful adjustments and regulation of the flow of gas and liquid, controlling in this way the temperature in the cooler.

Testing of Refrigerating Plants. The primary object of a test of a refrigerating apparatus is to compare the refrigerating effect with the heat equivalent of the mechanical work and of the cooling water or brine. The making of ice is not satisfactory for accurate results in a test. The range of temperature should not be greater than necessary to secure accuracy in the thermometer readings. A range of from 5 to 6 degrees Fahrenheit is usually sufficient and, in fact, most satisfactory. The brine should be measured or weighed in suitable tanks as for the condensed steam in engine tests.

One of the most important precautions to be observed is to determine accurately the specific heat of the brine for the temperature range of the test. Small differences in its concentration and composition may produce a considerable variation in results. When a compresser and steam engine are coupled directly together on the same shaft a direct measurement of the power required for the compresser is not obtainable. By measuring the horse power of the engine running without doing any work in the compresser—that is, operating it "empty"—and by comparing the differences in power between the steam engine and compresser for wide variations of condenser pressure, the effective horse power required to drive the refrigerating machine can be determined with some degree of accuracy. On account of a great deal of external friction being included it is not very satisfactory to use for this calculation the horse power of the compressor as determined with an engine indicator.

The following arrangement of data for a test of a refrigerating plant given by Kent is very useful. It is arranged for tests by either the compression or the absorption type of apparatus:

Report of Test.

Reports intended to be used for comparison with the figures found for other machines will therefore have to embrace at least the following observations:

Refrigerator:

Quantity of brine circulated per hour ....

Brine temperature at inlet to refrigerator, t ....

Brine temperature at outlet of refrigerator ....

Specific gravity of brine at 64 degrees Fahrenheit ....

Specific heat of brine ....

Heat abstracted (cold produced), Q_e ....

Absolute pressure in the refrigerator ....

Condenser:
- Quantity of cooling water per hour ....
- Temperature at inlet to condenser ....
- Temperature at outlet to condenser, t_c ....
- Heat abstracted, Q_2 ....
- Absolute pressure in the condenser ....
- Temperature of gases entering the condenser ....

Compression Machine.

Compressor:
- Indicated work, L_c ....
- Temperature of gases at inlet ....
- Temperature of gases at discharge ....

Steam engine:
- Feed-water per hour ....
- Temperature of feed-water ....
- Absolute steam pressure at steam engine ....
- Indicated work of steam engine, L_e ....
- Condensing water per hour ....
- Temperature of condensing water ....
- Total sum of losses by radiation and convection, $\pm Q_3$ ....
- Heat balance ....

$$Q_e + AL_c{}^{1} = Q_1 \pm Q_3 \quad . \quad . \quad . \quad . \quad . \qquad (107)$$

Absorption Machine.

Still:
- Steam consumed per hour ....
- Absolute pressure of heating steam ....
- Temperature of condensed steam at outlet ....
- Heat imparted to still, Q'_e ....

Absorber:
- Quantity of cooling water per hour ....
- Temperature at inlet ....
- Temperature at outlet ....
- Heat removed, Q_2 ....

Pump for ammonia liquor ....
- Indicated work of steam engine ....
- Steam consumption for pump ....
- Thermal equivalent for work of pump, ALp ....
- Total sum of losses by radiation and convection, $\pm Q_3$ ....

Heat balance:

$$Q_e + Q'_e = Q_1 + Q_2 \pm Q_3 \quad . \quad . \quad . \quad . \quad . \qquad (108)$$

For the temperatures T and T_c at which heat is abstracted in the refrigerator and imparted to the condenser, it is correct to select the temperature of the brine leaving the refrigerator and that of the cooling water leaving the condenser, because it is in principle impossible to keep the refrigerator pressure

[1] The term A is the reciprocal of the mechanical equivalent of heat (778).

higher than would correspond to the lowest brine temperature, or to reduce the condenser pressure below that corresponding to the outlet temperature of the cooling water.

Professor Linde shows that the maximum theoretical efficiency of a compression machine may be expressed by the formula,

$$\mathbf{Q} \div (\mathbf{AL}) = \mathbf{T} \div (\mathbf{T}_c - \mathbf{T}), \quad . \quad . \quad . \quad . \quad . \quad (109)$$

in which $\mathbf{Q}$ = quantity of heat abstracted (cold produced);
$\mathbf{AL}$ = thermal equivalent of the mechanical work expended;
$\mathbf{L}$ = the mechanical work;
$\mathbf{A} = 1 \div 778$;
$\mathbf{T}$ = absolute temperature of heat abstraction (refrigerator);
$\mathbf{T}_c$ = absolute temperature of heat rejection (condenser).

If $\mathbf{u}$ is the ratio between the heat equivalent of the mechanical work $\mathbf{AL}$ and the quantity of heat $\mathbf{Q'}$ which must be imparted to the motor to produce the work $\mathbf{L}$, then

$$\mathbf{AL} \div \mathbf{Q'} = \mathbf{u} \text{ and } \mathbf{Q'}/\mathbf{Q} = (\mathbf{T}_c - \mathbf{T}) \div (\mathbf{uT}), \quad . \quad . \quad (110)$$

It follows that the expenditure of heat $\mathbf{Q'}$ necessary for the production of the quality of cold $\mathbf{Q}$ in a compression-machine will be the smaller, the smaller the difference in temperature $\mathbf{T}_c - \mathbf{T}$.

The following data sheet is used in parts by Denton[1]:

1. Average high ammonia pressure above atmosphere . . . .
2. Average back ammonia pressure above atmosphere . . . .
3. Average temperature brine inlet . . . .
4. Average temperature brine outlet . . . .
5. Average range of temperature . . . .
6. Lbs. of brine circulated per minute . . . .

6a. Specific heat of brine . . . .

7. Average temperature condensing water at inlet . . . .
8. Average temperature condensing water at outlet . . . .
9. Average range of temperature . . . .
10. Lbs. water circulated per minute through condenser . . . .
11. Lbs. water per minute through jacket . . . .
12. Range of temperature in jackets . . . .
13. Lbs. ammonia circulated per minute . . . .

[1] *Transactions American Society of Mechanical Engineers.* Vol. 12, page 356.

14. Probable temperature of liquid ammonia entrance to brine-tank ....
15. Temperature ammonia corresponding to average back pressure......
16. Average temperature of gas leaving brine tank....................
17. Temperature of gas entering compressor..........................
18. Average temperature of gas leaving compressor .. ..
19. Average temperature of gas entering condenser....................
20. Temperature due to condensing pressure..........................
21. Heat given ammonia:
    By brine per B.T.U. per minute.............................
    By compressor, B.T.U. per minute..........................
    By atmosphere, B. T. U. per minute.........................
22. Total heat received by ammonia, B.T.U. per minute...............
23. Heat taken from ammonia:
    By condenser, B.T.U. per minute.............................
    By jackets, B.T.U. per minute...............................
    By atmosphere, B.T.U. per minute............................
24. Total heat rejected by ammonia, B.T.U. per minute...............
25. Difference of heat received and rejected, B.T.U. per minute ....
26. Per cent of work of compression removed by jackets...............
27. Average revolutions per minute..................................
28. Mean effective pressure steam cylinder, lbs. per square in...........
29. Mean effective pressure ammonia cylinder, lbs. per square in.......
30. Average H.P. steam cylinder......................................
31. Average H.P. ammonia cylinder...................................
32. Friction in per cent of steam H.P................................
33. Total cooling water, gallons per minute, per ton ice-melting capacity per 24 hours...
34. Tons ice-melting capacity per 24 hours...........................
35. Lbs. ice-refrigeration effect per lb. coal at 3 lbs. per H.P. hour.........
36. Cost coal per ton of ice-refrigerating effect at $4 per ton ....
37. Cost water per ton of ice-refrigerating effect at $1 per 1000 cu. ft....
38. Total cost of 1 ton ice-refrigeration effect.......................
39. Refrigeration effect per I.H.P. in compress. cyl., B.T.U. per minute..
40. Refrigeration effect per I.H.P. in steam, cyl., B.T.U. per minute....
41. Refrigeration effect per pound of steam, B.T.U. per minute.........

CHAPTER XVII

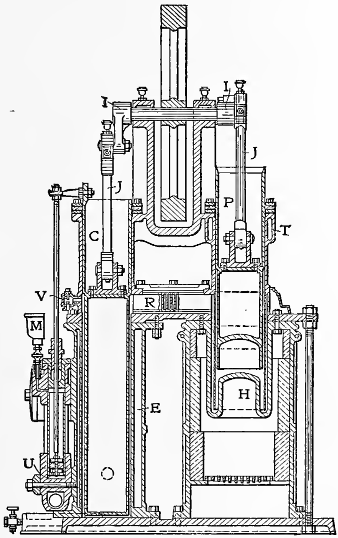
TESTING OF HOT-AIR ENGINES

Hot-air Engines of the conventional type are reciprocating "piston" engines which are operated by the alternate expansion and contraction of a charge of air. This alternate expansion and contraction is produced by heating and cooling. Engines of this kind now in use are found most often in country places, where they are used for pumping water. Usually coal is burned for fuel; but sometimes gas is used, particularly in the natural-gas districts.

Rider Hot-air Engine. The most successful engine of this kind is made by the Rider-Ericsson Engine Company of New York. This engine, illustrated in **Fig. 232,** consists of a compression cylinder **C** and a power cylinder **P**, each provided with a separate piston. These two cylinders are connected together by a rectangular passage **R** containing a large number of thin metallic plates and forming what is called in engines of this type the **regenerator.** This regenerator has for its function the alternate abstracting and returning to the air of a quantity of heat. Air leaking out is replaced by a fresh supply admitted through the check valve **V,** which opens inward. The compression cylinder **C** is provided with a water-jacket.

The cycle of operations in this engine consist of a compression stroke when the piston in the compression cylinder **C** compresses the cold air from which the heat has been abstracted by its passage through the regenerator **R,** and then by the simultaneous advancing upward movement of the piston in the power cylinder **P** the air passes again through the regenerator and also through the **heater H** without appreciable change of volume. As a result the addition of heat increases the pressure of the air and when it enters the power cylinder **P** it pushes the piston upward to the end of its stroke. This upward movement of

the power piston in the last half of its stroke carries with it the piston in the compression cylinder **C**, which is on the same shaft but set at an angle of 90 degrees, so that the two pistons do not reach the ends of their strokes together. Now as the charge of air cools the pressure falls, so that the piston in the power cylinder falls and in the last half of this stroke carries downward with it the piston in the compression cylinder and again starts compressing the charge of air. As the heated air passed through the regenerator plates on its way to the compression cylinder the greater portion of the heat it contained was left in them to be abstracted on the return movement to be used again for increasing the temperature of the charge.



C–Compression Cylinder
P–Power Cylinder
E–Cooler
H–Heater
R–Regenerator
II–Cranks set at about 100°
JJ–Connecting Rods
L–Check Valve
M–Pump Primer
T–Water Jacket, to protect packing from heat
U–Pump

FIG. 232. Hot Air Engine.

In **Fig. 232** a water pump **U** is shown at the left-hand side of the engine. Besides being used for supplying a water system it pumps the cooling water needed for the water-jacket **T** and the cooler **E**.

Tests of Hot-air Engines do not differ in the important details from tests of steam and gas engines. The indicated horse power is obtained by attaching engine indicators to both the power and the working cylinders and the net indicated horse power is the difference between that for the power and that for the compression cylinders. A Prony brake or similar device attached to the main shaft for absorbing the power can

be used to determine the useful or brake horse power and the ratio of the brake to the indicated horse power is the mechanical efficiency.

For testing such engine to determine the efficiencies and economy it is preferable to use gas or oil for fuel instead of coal, because of the obvious advantages in the determination of fuel consumption.

Therdomynamic efficiency is the ratio of the range of temperature to the initial absolute temperature of the air in the power cylinder. Temperatures not determinable by direct measurement may be calculated from the pressures and the specific volumes by the general formula for perfect gases,

$$T = pv/R, \qquad (111)$$

where R for air is 53.21.

CHAPTER XVIII

TESTS OF HOISTS, BELTS, AND FRICTION WHEELS

Efficiency of Hoists. An efficiency test of a hoist is made by determining the ratio between the work done in lifting the load to that applied to the hand chain. Stated briefly, this determination is made by raising slowly a known weight, observing at the same time by means of a spring balance fastened to the hand chain, the pull or force required to keep the load moving after it has been started. The method is, of course, the same for determining the efficiency of a rope hoist, except that some special provision must be made for attaching the hook of the spring balance to the rope. Allowance must also be made, of course, for the number of times the power is multiplied, that is, the relative velocity value.

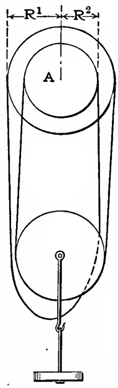


FIG. 233.—Differential Hoist.

Differential Hoists. Differential hoists (**Fig. 233**), are a little more complicated than the ordinary chain or rope hoist. In this apparatus, as shown in the standard books on the theory of mechanism, the velocity ratio is expressed according to the dimensions in the figure by,

$$\frac{2R_1}{R_1 - R_2}. \qquad (112)$$

It is difficult, however, to measure accurately the radius of these wheels on account of the irregular surface made for gripping the links. Now since the circumferences of these wheels are proportional to the radii, the velocity ratio may be determined by counting the number of link-pockets in each of the wheels, and its value will be given by the ratio of twice the number of

link-pockets in the larger wheel divided by the difference between the number of link-pockets in the larger and the smaller wheels. In some other types of hoists where this method is not applicable and the diameters cannot be readily measured, the velocity ratio can be determined by tying a piece of string on a link of the "load" chain or rope, as the case may be, opposite some fixed part of the hoist and mark in the same way a point on the chain or rope to which the pull is applied. Now when the "load" chain has been moved a measured distance, the corresponding movement of the point of application can be measured. Several observations should be made to eliminate probable errors in the measurements. Velocity ratio is usually expressed by making the second member of the ratio unity, thus 13:1, 4:1, etc.

The force required to move the hand chain by the spring balance multiplied by the velocity ratio is the work "put into" the hoist, while the weight lifted is proportional to the work done. Efficiency is then the work done divided by the work "put in," or in the terms above is the weight lifted divided by the product of the pull on the hand chain times the velocity ratio.[1] Determinations should also be made when the load is being lowered, but these results should not be averaged with those for raising the load because a hoist is generally used only for raising loads.

Determination of Tension in Belts and Rope Drives. Tests are often required to determine the power transmitted by belts and ropes for specified conditions of load, speed, tension, and the coefficient of friction between these and the pulleys on which they run. Suitable apparatus for such tests consists of a device with which the belts or ropes can be operated with different tensions. Power delivered to the follower is generally measured with a Prony brake. Usually the load is not increased beyond the limit producing 3 per cent slip.[2] The initial tension in the belt or rope should be measured when at rest. The total tension in motion is the sum of the tensions on the two

[1] If the spring balance is used in the inverted position, its weight must be added to the pull. Also the weight of the scale pans used for supporting the weights, must be added.

[2] Slip in bolts or ropes is ratio of the difference between the revolutions of the driver and the follower divided by the revolutions of the driver, each taken, of course, for the same time unit.

sides of the pulley $(T_1 + T_2)$. Now if T_1 and T_2 are equal there will be no motion of the belt or rope, but if T_1 is greater than T_2 the motion and the power transmitted will be proportional to the difference between $(T_1 - T_2)$ which is approximately expressed by the formula:

$$T_1 - T_2 = \frac{2(\text{length of brake arm (inches)} \times \text{net load on brake})}{\text{diameter of driven pulley (inches)}} \qquad (113)$$

When $T_1 + T_2$ and $T_1 - T_2$ are both known the separate tensions, T_1 and T_2 are, of course, readily calculated.

Tests of Friction Wheels. The apparatus frequently used for determining the **coefficient of friction** between friction wheels consists of a pair of pulleys, one of them at least usually made of some soft metal like aluminum or a fibrous material like straw fiber, leather fiber or paper. This driver runs on a follower generally of some other material. Power delivered to the "follower" shaft is absorbed by the Prony brake. A bell-crank lever to which weights can be attached is used to hold or press the two pulleys together.

The coefficient of friction as determined by such an apparatus is the ratio of the tangential pull to the total normal pressure. If the coefficient of friction is represented by **f**, and other symbols are used as follows:

r_1 = the effective brake arm in inches;
r_2 = the radius of driven pulley;
w = net weight on the brake in pounds;
p = normal pressure in pounds per inch of width;
z = width of the narrower pulley in inches.

Then,

$$f = \frac{\frac{wr_1}{r_2}}{pz} = \frac{wr_1}{pr_2z}. \qquad (114)$$

CHAPTER XIX

HYDRAULIC MACHINERY

Tests of Hydraulic Machinery

Tests of Boiler Feed-pumps. In general engineering practice there are two classes of pumps commonly used for "feeding" the water to steam boilers. These are:

(1) Motor- or belt-driven pumps;

(2) Steam pumps.

Motor-driven feed-pumps are more generally used in Europe than in America, but there are, however, many plants in which feed-pumps operated by direct connection to an electric motor or by belting from a line shaft are used. **Fig. 236** shows a good modern example of such a pump. It has three plungers operated from a single shaft with the cranks set 120 degrees apart. The valves are accessible for cleaning or for repairs by removing the plates or "covers" **C, C, C.** The suction pipe is marked **S** and the discharge pipe **D.** An air-chamber **A** is provided to produce, by cushioning air in it, a somewhat more steady flow than would be secured without it. A **relief valve** should be provided on the discharge pipe to act as a safety valve in case the pressure in the line should get so high that the pump itself might be broken.

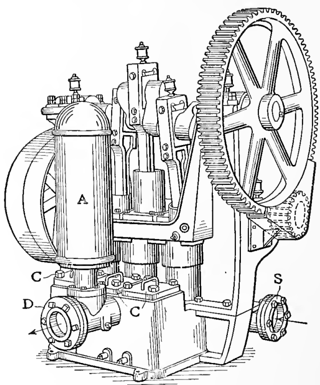


Fig. 236.—Belt-driven Feed Pump.

The power delivered to a belt-driven pump can usually be conveniently measured with a Webber dynamometer or an Emerson power scales (see pages 136–140), while if it is direct-connected to a motor the efficiency of the motor may be obtained by disconnecting it, attaching a Prony brake to the shaft, and measuring the input to the motor with suitable electrical instruments.

The **equivalent work done "on the water"** by the pump is found by multiplying the total head [1] (suction + discharge) in **feet** by the weight of water lifted (foot-pounds).

The quantity of water delivered can be determined by weighing or by calculating the flow over a weir or from an orifice (see pages 156–161).

Slip is the difference between the volume swept through by the plunger of the pump; or, in general, the piston displacement, and the actual volume of the water pumped at the required head.

In piston pumps, direct-connected to the steam cylinder without a crank, the length of the stroke is usually variable, and some special method must be adopted for such tests to determine the average length of the stroke.

Duty of a pump is usually defined as the number of foot-pounds of work "delivered" by the pump per 1,000,000 B.T.U. supplied. The heat units supplied by the engine are calculated by the A.S.M.E. Rules [2] as the product of the weight of feed-water used by the boiler and the total heat of steam at boiler pressure "reckoned from the temperature of the feed-water." The total heat is to be corrected of course for moisture or superheat.

For a test utilizing a Webber or a similar dynamometer belted to the pump for measuring the power the following form may be used:

[1] If the **discharge** head is measured by a pressure-gage on the discharge pipe then the equivalent pressure in pounds per square inch corresponding to the difference in level between the surface of the water-supply and the center of the gage must be added to get the **total** head. (One foot head of water at about 62 degrees Fahrenheit is equivalent to 0.434 pound per square inch; and conversely, one pound per square inch is equivalent to a head of 2.305 feet of water at the above temperature. See also foot-note page 361.

[2] More detailed instructions for steam pumps with steam jackets are given in *Transactions American Society of Mechanical Engineers*, vol. 12, page 530.

Test of a Belt-driven Pump
(Dynamometer Method)

1. Type of pump................ Made by.......................
2. Diameter of plungers, ins..
3. Length of stroke, ft..
4. Size of suction pipe..
5. Size of delivery pipe...
6. Speed of dynamometer, r.p.m......................................
7. Speed of pump, r.p.m..
8. Dynamometer reading..
9. Delivery pressure, lbs. per sq.in.................................
10. Suction pressure, lbs. per sq.in. or inches vacuum.....................
11. Temperature of water, deg. F.....................................
12. Delivery head in feet of water....................................
13. Suction head in feet of water.....................................
14. Total head in feet of water...
15. Net weight of water pumped per minute, lbs.....................
16. Work done by pump, ft.-lbs. per min. (14)×(15)................
17. Cubic feet water pumped per minute.............................
18. Plunger displacement, cu.ft. per min............................
19. Slip, per cent [(18)−(17)]÷(18)......................................
20. Net work delivered to pump (by dynamometer) ft.-lbs. per minute..
21. Dynamometer horse power, (20)÷33,000..........................
22. Pump horse power, (16)÷33,000....................................
23. Mechanical efficiency, (22)÷(21)..................................
24. Capacity of pump, gallons delivered per 24 hours.................

A Direct-acting Steam Feed-pump like the one shown in section in **Fig. 237** will be tested in a somewhat different manner, and a different set of observations is required.

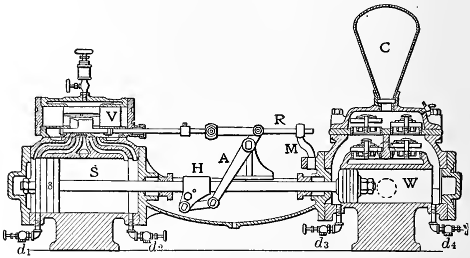


Fig. 237.—Direct-acting Steam Pump.

In none of the so-called direct-acting steam pumps has a rotary motion been developed by means of which an eccentric can be made to operate the valve. It is, therefore, necessary to reverse the piston by an impulse derived from itself at the end of

each stroke. This cannot be effected in an ordinary single-valve engine, as the valve would be moved only to the center of its motion, and then the whole machine would stop. To overcome this difficulty a small steam piston is provided to move the main valve of the engine.

In these pumps, the lever **A,** which is carried by the piston rod, comes in contact with the tappit when near the end of its motion, and, by means of the valve rod **R** moves the small slide valve which operates the supplemental piston. The supplemental piston, carrying with it the main valve **V,** is thus driven over by steam, and the engine reversed. If, however, the supplemental piston fails accidentally to be moved, or to be moved with sufficient promptness by steam, the lug on the valve

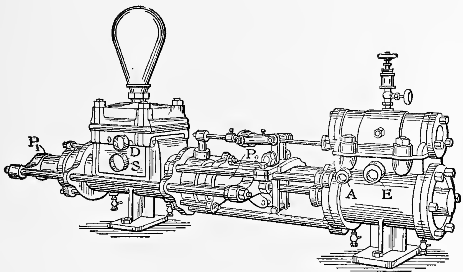


FIG. 238.—Outside Packed Plunger Feed-pump.

rod engages with it and compels its motion by power derived from the main engine.

Outside-packed steam pumps of the plunger type (**Fig. 238**) are now very commonly used for supplying boiler feed-water, chiefly because at the pump end the only part subjected to ordinary wear is the packing of the plunger stuffing-boxes. Steam is admitted at **A** and exhausts at **E.** The suction pipe is at **S** and the discharge pipe at **D.**

Suitable fittings for the attachment of indicators should be provided at both the steam and the water cylinders. If the pump is of the ordinary direct-connected type, without a fly-wheel, like the one shown in **Fig. 237,** some provision must be made to make regular observations of the **length** of the stroke,

as it is scarcely ever constant. One method is to attach a suitable arm to the cross-head **H, Fig. 237,** with a pencil at the end. Strips of tough paper can then be pasted on a board in such a position that the pencil will trace the lengths of the strokes. By shifting the position of the board every minute or two, records will be obtained from which the average length of the stroke can be estimated with considerable accuracy. Another method giving still greater accuracy is to use a counting device designed by Professor Cooley, operated by a mechanism similar to that in a clock (**Fig. 240**). A cord from the instrument is attached to the cross-head of the pump and the clock mechanism moved by this cord integrates or sums the lengths

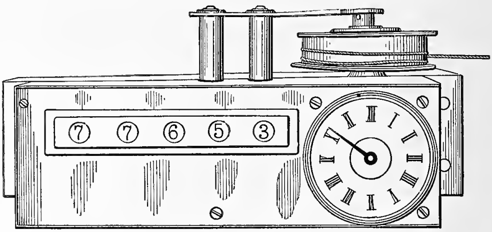


Fig. 240.—Cooley Stroke-measuring Devise.

of all the strokes. Conditions should be maintained constant before beginning a test.

Form for Steam Pump Test

1. Duration of test. . . . .
2. Diameter of steam cylinder. . . . .
3. Diameter of piston rod. . . . .
4. Diameter of water plunger. . . . .
5. Diameter of plunger piston rod. . . . .
6. Displacement of plunger, cu. ft. { Head end. . . . .
   Crank end. . . . .
7. Average length of stroke, ft. . . . .
8. Average number of strokes per minute. . . . .
9. Temperature of water, deg. F. . . . .
10. Temperature of feed-water to boiler, deg. F. . . . .
11. Temperature in steam calorimeter, deg. F. . . . .
12. Feed-water supplied to boiler, lbs. per hour. . . . .
13. Quality of steam. . . . .
14. Dry steam supplied to boiler, lbs. per hour. . . . .
15. Boiler pressure, lbs. per sq.in. . . . .
16. Delivery pressure, lbs. per sq.in. . . . .

17. Suction pressure, lbs. per sq.in. or inches vacuum.................
18. Vertical distance between top of suction pipe where suction gage is attached and center of gage on delivery pipe, ft.[1]..............
19. Total head in feet of water.......................................
20. Weight of water delivered per hour, lbs...........................
21. Plunger displacement, cu.ft. per hour.............................
22. Weight of water by plunger displacement, lbs. per hour............
23. Slip of pump, per cent [(21) − (19)] ÷ (21).......................
24. Coal fired per hour, lbs..
25. Combustible burned per hour, lbs..................................
26. Steam used per pound of coal, lbs. (actual evaporation)...........
27. Equivalent evaporation ("from and at 212 deg. F.") per pound of of coal, lbs..
28. Actual evaporation per pound of combustible, lbs..................
29. Equivalent evaporation per pound of combustible, lbs..............
30. Duty, per 1,000,000 B.T.U...
31. Duty, per 100 lbs. coal fired.....................................
32. Duty, per 1000 lbs. steam (dry)...................................
33. Capacity, gallons delivered per 24 hours..........................
34. Mean effective pressure, steam cylinders, lbs. per sq.in..........
35. Mean effective pressure, water cylinders, lbs. per sq.in. ....
36. Indicated horse power, steam cylinders............................
37. Indicated horse power, water cylinders............................
38. Dry steam used per indicated horse power per hour (steam cylinders)

Cooley Stroke-measuring Device. An apparatus has been developed at the University of Michigan for measuring accurately the length of the stroke of the type of pumps in which steam and water cylinders are direct-connected on the same piston rod, such for example as the ordinary steam feed-pumps. In such pumps it scarcely ever happens that there are two strokes in succession that are of the same length and more or less approximate methods are usually adopted for obtaining the average length of the stroke during a test. With the stroke-measuring device referred to above each individual stroke is accurately measured and is added by a counting device to the sum of all the other strokes that have preceded. As this apparatus is used in a test of a variable stroke pump, the reading of the counter to the nearest inch can be recorded at the usual times for observations. The difference between two readings is the total length of all the strokes for the interval between observations. If, then, this difference is divided by the total

[1] This is usually stated as the vertical distance between the two gages. A vacuum gage, however, on the suction pipe of a pump indicates the vacuum at the level of the *top of the suction pipe*, and not up to the center of the gage. This was shown by Professor Cooley by attaching a gage by means of a suitable fitting to the suction pipe of a pump so that the gage could be revolved above and below the pipe. It was observed that the reading of the gage remained constant, showing that in a suction pipe of a pump the water does not rise higher than the top of the pipe.

number of strokes for the same time, the average length of the stroke can be determined accurately. An assembled view of this device is shown in **Fig. 240,** and its mechanism is shown in **Fig. 241,** which it will be observed is the same in principle as the silent ratchet clutches used for the continuous indicator described on page 100. The apparatus is driven by the cord on the wheel **W,** which moves the ratchet wheels **B** and **C** in the same way as

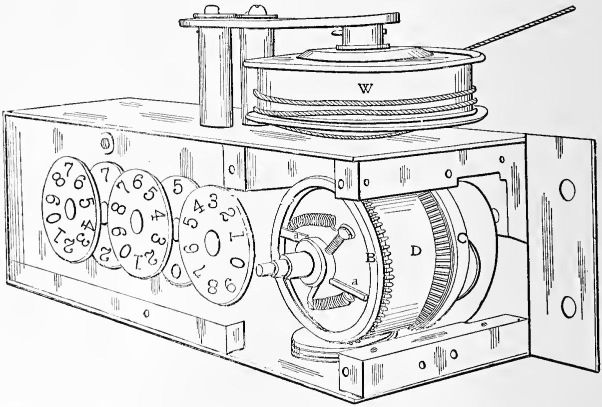


FIG. 241.—Mechanism of Stroke-measuring Device.

the corresponding parts are moved in the continuous indicator referred to. Numbers on the horizontal plate (**Fig. 240**) are **feet** and those on the circular dial are **inches.**

Tests of Centrifugal Pumps. Tests of pumps operating against low heads such as single-stage centrifugal pumps are suitable for, are made in the same way as explained, for the triplex belt-driven feed-pump. It is desired, of course, from the results of the tests to compare the power supplied to the pump with the work done in lifting the water. Power supplied would probably be again measured by some form of transmission dynamometer, and the work done is calculated from the weight of water delivered and the total head against which the pump delivers.[1]

[1] For more detailed testing of centrifugal pumps see "Centrifugal Pumps," by Lowenstein and Crissey (D. Van Nostrand Co., 1911).

Centrifugal pumps are frequently driven by direct-connected steam turbines. The horse power required to drive the pump is then determined from a speed-power curve of the turbine (**Fig. 203**), page 281, obtained usually from a Prony brake test of the turbine. Similarly, if the pump is driven by a variable-speed electric motor, a speed-power curve of the motor can be used. Usually, however, when a constant speed motor is used it is simpler to determine an efficiency curve of the motor for varying power.

Tests of Impulse Water Wheels. Impulse wheels used to operate with water under pressure consist usually of a series of buckets attached to the periphery of a disk or wheel. The buckets are usually divided by a central rib so that two "pockets" are formed (**Fig. 242**). The curves for each of the divisions of the bucket are designed to turn the direction of the impinging steam without shock. **Fig. 243** shows a typical impulse wheel. Impulse wheels are designed to operate most efficiently with high heads. It is, therefore, impracticable to measure the head directly in feet, but it is done usually by measuring the pressure near the nozzle **N** with a gage. When the center of the gage is at a higher level than the center of the nozzle discharging on the wheel, then this difference in level must be added to the head calculated from the gage pressure to determine the total head under which the wheel is operating. Power developed is measured usually by a Prony brake connected to the shaft **S.** In all tests where a large quantity of water is used, the temperature of the water should be recorded and the weight corresponding

Fig. 242.—Bucket of an Impulse Water Wheel.

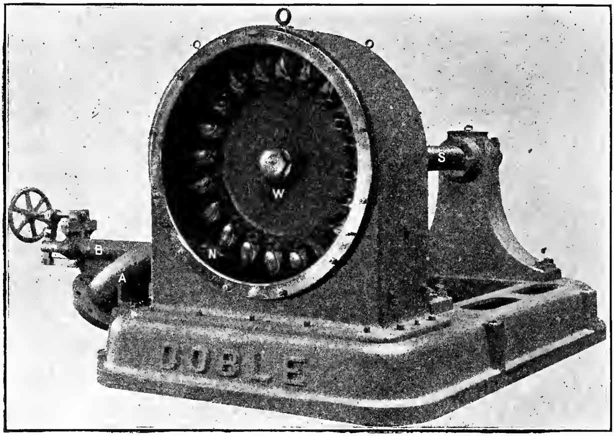


FIG. 243.—Typical Impulse Water Wheel.

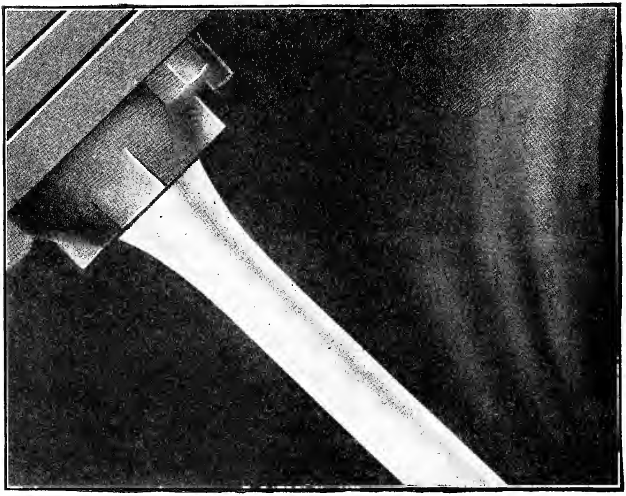


FIG. 244.—Water Jet Discharging at High Pressure from the Nozzle of an Impulse Wheel.

should be used. A view of the jet discharged from one of these nozzles is given in **Fig. 244.** The type of impulse wheel most in use commercially is called the Pelton, of which typical buckets and the engaging jet of water are shown in **Fig. 245.**

Laboratory tests for a **given head** are usually run when varying both the load and speed. Make the first test with the load on the Prony brake as light as possible consistent

FIG. 245.—Buckets and Jet of a Pelton Wheel.

with fairly steady operation of the wheel, and then take a series of tests increasing the load in increments to reduce the speed about 100 revolutions per minute in each succeeding test. Duration of tests at each speed should be from twenty to thirty minutes with observations taken every two minutes. The following from may be used for tests:

TEST OF IMPULSE WHEEL

General Data:

1. Date..
2. Name of wheel and nominal horse power ....
3. Kind of bucket..
4. No. of buckets..
5. Angle of buckets..
6. Diameter of bucket wheel, inches ....
7. Area of nozzle and delivery pipe........................
8. Coefficient of discharge for type of nozzle........................
9. Diameter of brake wheel, inches ....
10. Length of brake arm, inches ....
11. Tare of brake, lbs..
12. Duration of test..
13. Average temperature of water, deg. F........................

14. Average pressure by gage at wheel, lbs. per sq.in..................
15. Average head at wheel in feet[1]......................................
16. Quantity of water for total run in pounds.........................
17. Quantity of water in pounds per minute.........................
18. Cubic feet of water per minute.....................................
19. Foot-pounds of work per minute calculated from (15) and (17)......
20. R.P.M..
21. Net weight on brake, lbs..
22. Horse power as measured by brake...............................
23. Over-all efficiency of motor, per cent (22) ÷ (19) × 33,000...........

[1] Corrected for vertical distance from the center of the gage to the center of the nozzle.

Curves. Plot a curve for each head with speed for abscissas and efficiency per cent for ordinates, also curves for the ratio of the velocity of the periphery of the wheel $\mathbf{v}_p$ to the theoretical velocity due to the head $\mathbf{v}_t$; that is, $\mathbf{v}_p/\mathbf{v}_t$ for abscissas and the maximum horse power developed for ordinates.[1]

Tests of Water Turbines. A typical reaction turbine is shown in **Fig. 246.** The power is transmitted by the main shaft and the smaller shaft is used for controlling the gates regulating the quantity of water passing through the wheel. For testing, a Prony brake can be placed directly on the vertical shaft. In some respects a rope brake is most suitable, as one end can be attached to a spring balance and the other end can be led over a pulley, and will thus support weights on a vertical hanger. It is preferable to have the lower web of the brake-wheel solid, that is, without arms, so that it will retain the cooling water, which should be arranged to flow into it at the rate required.

Power supplied is determined by the weight of water used and by the head under which the wheel operates. These quantities are determined in the same general way as for a test of an impulse wheel, already described, **except in the case of a reaction turbine,** where the housing or casing in which the wheel is placed is **always completely filled with water (Fig. 247).** With this arrangement the turbine receives not only the effect due to the pressure-head, measured from the level in the head-race to the center of the wheel, **but also that due to the suction head,** measured from the center of the wheel to the level in the tailrace. Data can be recorded in a form similar to that for

[1] Plot curves showing effect of head on efficiency if several tests are run at different heads.

tests on impulse wheels. A typical runner for a reaction turbine is shown in **Fig. 248.**

Curves. Plot a curve for each gate opening at a constant head with speed for abscissas and efficiency per cent for ordinates.

Tests of Hydraulic Rams. A section of a typical hydraulic ram is shown in **Fig. 249.** It consists of an air chamber **H,**

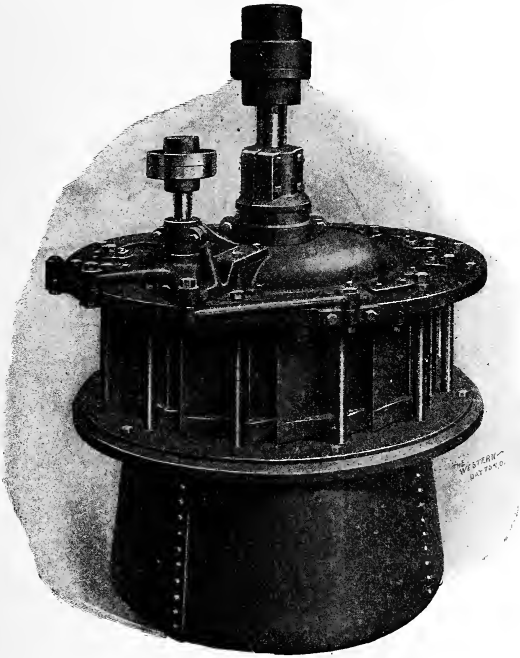


FIG. 246.—Typical Reaction Water Turbine.

to which is connected the discharge pipe **I.** There is a check valve **G** opening into the air chamber from the lower chamber **A** into which water is brought by the pipe **S.** There is a waste valve at **B.** This valve is weighted and opens inward. By means of a nut **J** on the stem of this valve the lift or amount of opening of the valve can be regulated. When water is supplied

to the ram, it escapes through the waste valve **B** with a velocity corresponding approximately to that due to the head under which the water is supplied. The effect of this velocity head is to reduce the pressure on the upper side of the valve so that it becomes unbalanced and closes suddenly. Then the momentum of the column of water in the pipe **S** becomes sufficient to open

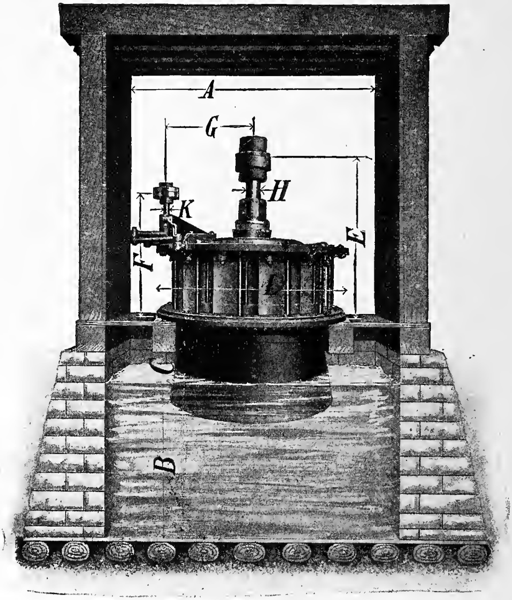


FIG. 247.—Reaction Turbine with Submerged Housing.

the valve **G,** and will discharge some water into the discharge pipe **I** against a considerable head. As soon as the pressures become equalized the valve **G** closes, the waste valve **B** opens and water from the supply pipe is again "wasted." This alternate action is produced with regularity, and as a result the water in the supply pipe acquires a certain "backward and forward" wave-motion. As the rule is generally stated the length of the supply pipe leading from the reservoir to the ram

must be at least five times the head. This length is necessary to secure some resistance to this " backward and forward " wave-motion. A small air chamber shown at **P**, with a check-valve **C** opening inward to supply air, is provided in many of these rams, as it improves the efficiency. The rate of opening of the waste valve or the number of pulsations in a given time can be varied by changing the weight on its stem. This appara-

FIG. 248.—Typical Runner of a Reaction Turbine.

tus is tested usually by measuring the supply and the discharge heads, the weight of water discharged through the delivery pipe $\mathbf{w}_1$ and that passing through the waste valve $\mathbf{w}_2$ in pounds per minute. Then the available energy in the water is $(\mathbf{w}_1 + \mathbf{w}_2)\mathbf{h}_s$, where $\mathbf{h}_s$ is the supply head; and the useful work is $\mathbf{w}_1\mathbf{h}_d$, where $\mathbf{h}_d$ is the discharge head,[1] then,

$$\textbf{Efficiency} = \frac{\mathbf{w}_1\mathbf{h}_d}{(\mathbf{w}_1 + \mathbf{w}_2)\mathbf{h}_s}. \quad . \quad . \quad . \quad . \quad (115)$$

[1] Both the supply and the discharge heads must be measured, of course, from the same datum or " zero " level.

and the **capacity Q** in gallons per twenty-four hours is $\mathbf{Q} = \mathbf{1440\ w_1 q}$, where **q** is the fraction of a gallon of water in a pound. Satisfactory runs of twenty minutes' duration can usually be made, each run being made with a different lift or "stroke" of the waste valve **B.** Observations of the heads should be taken every five minutes if they are variable, and weighing as often as necessary, depending on the size of the tanks used. The effect on the efficiency of increasing the lift or "stroke" of the waste valve from one-eighth inch by increments of one-eighth inch is very interesting.

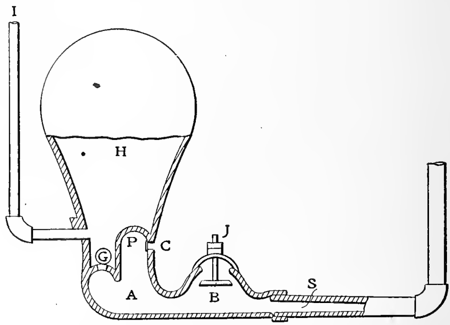


Fig. 249.—Section of a Simple Hydraulic Ram.

Curves. Plot curves with length of "stroke" as abscissas and take for ordinates:

(1) Efficiency;

(2) Capacity in gallons per twenty-four hours;

(3) Strokes per minute.

Fig. 250 shows a slightly different form of ram, as made commercially. The principle of operation is, however, the same as the one in **Fig. 249.** Letters used for marking the parts are the same in the two figures.

Tests of Pulsometers. A type of steam pump called a pulsometer is illustrated in section in **Fig. 251.** In the form shown here it consists of two chambers **AA,** joined by tapering necks into which a ball **C** is fitted so as to move in the direction of the least pressure between seats in these tapering passages.

The chambers **AA,** on opposite sides, are connected by means of check or clack valves **EE** with the "induction" chamber **D.** Water is delivered through the passage **H,** which is connected to the chambers through the valves **G.** Between the chambers is also a vacuum chamber **J,** connecting them with the "induction" chamber **D.** Small air valves, moving inward, supply air to the chambers **AA** by opening when the pressure is less

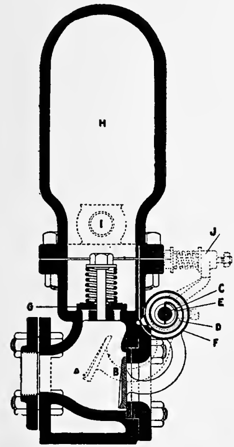


FIG. 250.—Commercial Type of Hydraulic Ram.

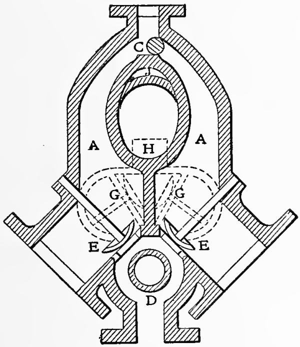


FIG. 251.—Steam Pulsometer.

than atmospheric. Its operation is explained briefly as follows: Starting with the left-hand chamber full of water and with a vacuum in the right-hand chamber, this latter chamber will fill with water which by its momentum due to rushing in suddenly, pushes back the valve **C** toward the left. Now **during this time** steam has entered the left-hand chamber to the left of the valve **C** (before it has shifted) and by exerting a pressure on the surface of the water it forces it through the check valve **G,** first into the delivery passage **H** and then into the air chamber **J.** Then the steam in this left-hand chamber condenses in contact with the cold water and forms a vacuum, permitting the repeti-

tion of this cycle of events, except that the operations in the two chambers are reversed.

Since all the steam used is condensed and discharged with the water lifted, the analysis of the operations in a pulsometer are similar to those in the familiar types of injectors, except that the steam acts in the pulsometer by pressure instead of by impact as in the injector.

Using the following symbols:

w_s = weight of dry steam, pounds;
w_w = weight of water lifted, pounds;
t_1 = temperature of the water supply, deg. Fahr.;
t_2 = temperature of the water delivered, deg. Fahr.;
r = the latent head of evaporation of the steam in B.T.U.;
t_s = the temperature of the steam, deg. Fahr.;
h_1 = the **suction** head, feet;
h_2 = the **delivery** head, feet;
$h_1 + h_2$ = the total head, feet, then

$$w_s(t_s - t_2 + r) = w_w(t_2 - t_1). \quad . \quad . \quad . \quad . \quad (116)$$

The heat equivalent of the mechanical work done is in B.T.U.,

$$\frac{1}{778}(w_w h_1 + (w_s + w_w)h_2),$$

and the heat expended is in B.T.U.,

$$w_s(t_s - t_2 + r),$$

and

$$\textbf{Thermal Efficiency} = \frac{w_w h_1 + (w_s + w_w)h_2}{778(w_s(t_s - t_2 + r))}. \quad . \quad (117)$$

And if we neglect the work done in lifting the condensed steam,

$$\textbf{Efficiency} = \frac{h_1 + h_2}{778(t_2 - t_1)}. \quad . \quad . \quad . \quad . \quad . \quad (118)$$

Curves. Plot with discharge pressures for abscissas curves with both thermal efficiency and capacity (gallons per twenty-four hours) as ordinates.

Tests of Injectors. The injector is known particularly in stationary service as the device used for pumping water into the boiler when the feed-pump fails. One of the various forms of injectors sold commercially is shown in **Fig. 252.** The **steam**

supply, the **suction** or water supply, the **delivery** or discharge, and the **overflow** are marked clearly. A **double-tube** injector, shown similarly in section in **Fig. 253,** has the parts marked in the same way as in the preceding figure.

Method of Operating Injectors. The method to be given, although applicable particularly to the ones described, is, however, more or less generally applicable to all makes. Open **wide** both the **steam-** and **water**-supply (suction) valves. Then **close** the water-supply (suction) valve **slowly till the overflow ceases** (for the type shown by **Fig. 252**); or (for the type of **Fig. 253**) pull the starting lever back a short distance until water appears at the overflow and then continue the movement steadily as far as the lever will go. Regulate the rate of delivery by closing the water-supply (suction) valve. Before testing an injector or indeed even before trying to operate a new injector, inspect the pipe fittings and particularly the valves on the water-supply pipe to observe whether they are tight. It is not at all unusual to find that the valve is not air tight, and for this reason it is a very good practice to put always some new wicking in the space for packing around the stem of the valve on the water-supply pipe; and turn up the cap over the packing tightly.

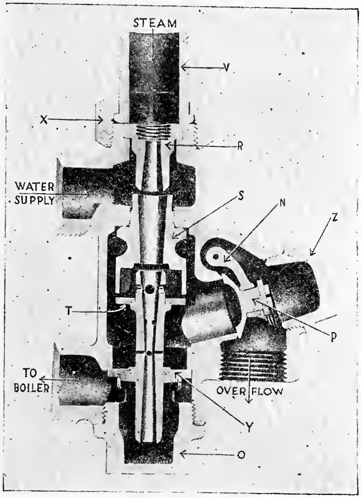


FIG. 252.—Single Tube Steam Injector.

Method of Testing. For the testing of injectors, the arrangement of apparatus consists usually of two barrels supported on platform scales, or carefully calibrated tanks fitted with gage glasses. During a test the injector draws water from one barrel

or tank and discharges it into the other. A test of an injector must be made, of course, with established conditions; that is, with a flying start. This may be accomplished by having the injector draw water from the supply tank, but discharge water through a by-pass connection on the discharge pipe till the test is to begin. For this preliminary operation of the injector the level in the supply tank can be maintained very closely, at any point marked by manipulating a "quick-opening" valve. When the test is to begin, close as quickly as possible this valve on the pipe discharging into the supply tank and turn the discharge from the by-pass into the delivery tank. To make this adjustment all the valves to be operated should be of the "quick-opening" type. The pressure against which the injector is to operate is secured by **throttling** the discharge pipe by means of a globe or an angle valve placed between the injector and the by-pass on the discharge pipe. The quick-opening valve would not be satisfactory. The suction head is measured from the middle of the injector to the **average** level of the water in the supply tank. The discharge head is obtained by adding to the head in feet corresponding to the pressure indicated by the gage the distance in feet from the center of the gage to a horizontal line through the middle of the injector. The temperatures of the water in the supply and delivery pipes must be observed. The injector is stopped at the end of the test by closing the steam valve.

The following form, similar to the one used at Purdue University, is very complete. Notes explaining the calculations required are given above:

TEST OF AN INJECTOR

Make of injector.............................. Date..............
Number..
Size of connections: steam........in. dia.; water............in. dia.;
discharge........in. dia.; area of discharge ($=a$)..........sq.in.
Diameter (minimum) of lifting tube......in.; forcing tube........in.

a. Duration of test..
b. Steam pressure (average) pounds gage, p_s........................
c. Delivery pressure (average) pounds gage, p_2......................
d. Maximum pressure against which injector will discharge, p_{max}.......
e. Suction-head (average), feet, h_1................................
f. Delivery-head (average), feet, h_2................................
g. Temperature of supply (average) t_1..............................
h. Temperature of delivery (average) t_2.............................
i. Pounds water supplied per hour, w_w..............................

j. Pounds water and steam delivered per hour, w_m ....
k. Cubic feet of water delivered per hour, Q ....
l. Wet steam per hour, $w_s(=w_m-w_w)$ ....
m. Dry steam per hour, $w'_s(=xw_s)$ ....
n. Water delivered per pound wet steam, pounds $(=w_w \div w_s)$ ....
o. Water delivered per pound dry steam, pounds $(=w_w \div w'_s)$ ....
p. Velocity of discharge, feet per second, $v(=144Q \div 3600a)$ ....
q. Energy delivered, raising injection water, B.T.U. per hour ....
r. Energy delivered, heating injection water, B.T.U. per hour ....
s. Energy delivered, velocity of discharge, B.T.U. per hour ....
t. Total energy delivered, B.T.U. per hour ....
u. Energy supplied, B.T.U. per hour ....
v. Thermal efficiency as a boiler-feed apparatus ....
w. Thermal efficiency as a pump ....
x. Horse power ....
y. Dry steam per horse power per hour, pounds ....

The energy of *raising injection water* $=[w_w(h_1+h_2)+w_sh_2]\div 778$, B.T.U. per hour.

The energy of *heating injection water* $=w_w(q_2-q_1)$ where q_1 and q_2 correspond to t_1 and t_2., B.T.U. per hour.

The *energy of discharge* $=w_mv^2 \div (2g\times 778)$, B.T.U. per hour.

The total energy delivered = item q + item r + item s.

The energy supplied $=w_s(xr_s+q_s-q_2)$, where r_s and q_s correspond to p_s, and q_2 corresponds to t_2. x = quality of steam.

The thermal efficiency as a boiler feed apparatus $=100\times\dfrac{\text{item } t}{\text{item } v}$.

The thermal efficiency as a pump $=100\times\dfrac{\text{item } q+\text{ item } s}{\text{item } v}$.

$$\text{The horse power}=\frac{w_w(h_1+h_2)+w_sh_2}{60\times 33{,}000}. \qquad (120)$$

The dry steam per horse power, per hour $=w_s'\div$ item x.

$$\text{The pump duty}=\frac{1{,}000{,}000+\text{item } p}{\text{item } t}$$

$$=\frac{1{,}000{,}000[w_w(h_1+h_2)+w_sh_2]}{778w_s(xr_s+q_s-q_2)}. \qquad (121)$$

The **weight of steam** found by direct weighing may be **checked**, by calculating (assuming radiation loss negligible) a "heat balance" in which this weight will be the only unknown, thus for this condition,

$$w_s(xr_s+q_s-q_2)=\frac{1}{778}\left[w_w(h_1+h_2)+w_sh_2+(w_w+w_s)\frac{v^2}{2g}\right] + w_w(q_2-q_1),$$

or, approximately,

$$w_s=\frac{w_w\left[h_1+h_2+778(q_2-q_1)+\dfrac{v^2}{2g}\right]}{778(xr_s+q_s-q_2)-h_2}. \qquad (119)$$

CHAPTER XX

TESTING THE STRENGTH OF MATERIALS

Machines for Testing the Strength of Materials consist, in general, of (1) a **power system** for producing in the specimen tested the required stresses, and (2) a weighing system to determine the amount of power applied. In the usual form of testing machine the load is applied to the specimen through a train of gears and screws operated either by power or by hand, depending, of course, largely on the capacity. The stress is measured by balancing the force exerted on the specimen by a poise adjusted at the end of a system of levers just as weight is determined with a platform scales. The general principle of most machines for testing materials is illustrated in a simple form in the apparatus for calibrating indicator springs, **Fig. 99,** page 107. In this case the power is applied to the hand wheel, which exerts two forces equal but opposite in direction, one compressing the spring in the indicator and the other pressing on the platform of the scales.

A diagrammatic view of a typical machine for testing materials in tension and compression is shown in **Fig. 255.** It consists essentially of a table **T,** to which the upper "head" **A** is rigidly attached by means of the vertical bars **DD.** Heavy vertical screws **SS,** carrying the lower cross-head **B** are moved up or down by the system of gears **GG.** Moving the cross-head **B** downward puts a tensile stress on a test specimen **s** if it is attached firmly to both the upper "head" **A** and to the cross-head **B.** The force applied to the specimen is transmitted by the bars **DD** to the weighing table **T,** which rests on the first weighing lever **M,** having a fulcrum at **F.** The load on the table **T** is applied to the lever **M** at the middle of the table. The long arm of this lever is connected by means of a short link to a second lever **N,** and this again is connected to the short arm of the lever **Q** at the other end of which the weighing poise **P** is

to be balanced. The position of the poise on this last lever (scale beam) indicates the force applied to the specimen s.

Fig. 256 is a remarkably good illustration for showing the parts of a standard testing machine and for explaining its operation. Vertical screws **SS**, connected by gearing to the power system, move the cross-head **B** up or down according to the direction of motion. The speed of these screws is controlled and their motion reversed by manipulation of the levers marked

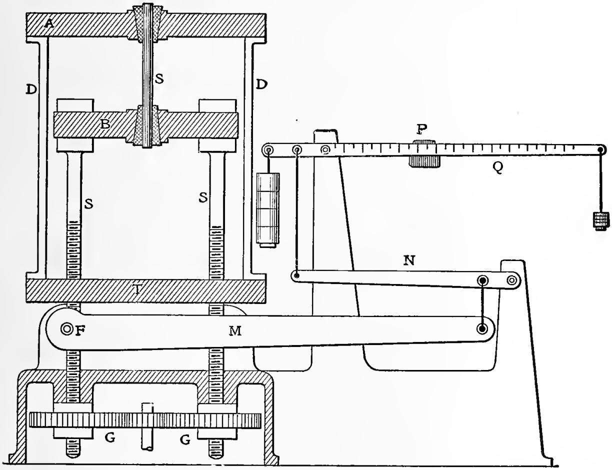


FIG. 255.—Diagram of a Simple Machine for Testing the Strength of Materials.

l_1, l_2, and l_3. The vertical columns supporting the upper "head" **A** are bolted to the table **T**, which rests on the system of levers **M, N, O,** and **Q**. The poise **P** is moved on the lever or scale-beam **Q** by means of a cord connected to the hand wheel **W**. The levers are balanced "to zero" by means of the counterpoise **C**. Adjustment for use with long or short specimens is secured by raising or lowering the upper head **A**. To prevent sudden jarring of the machine when the load is released by the breaking of a specimen, vertical rods fastened to the base pass up loosely through holes in the table **T**, at its four corners, and

on their ends large "**check-nuts**" are screwed. When the machine is in use these nuts must be loose, otherwise they will cause a pressure on the table causing the indication of the scale-beam to be greater than the weight due to the load on the specimen.

Small testing machines with a capacity not exceeding 50,000 pounds are made to operate by **hydraulic pressure.** In this

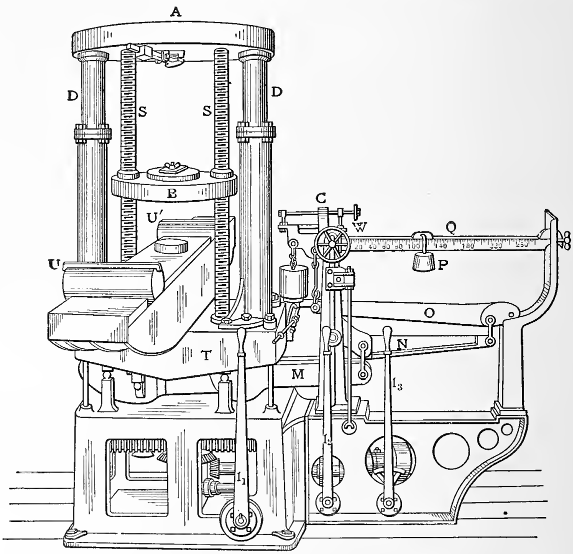


Fig. 256.—Standard Testing Machine.

type of machine the movable head applying the load to the specimen is moved by the pressure on a piston in a hydraulic cylinder. This hydraulic pressure is produced usually in a small hand-operated pump at the side of the machine. Oil is generally used for the working medium. In order to return the oil to the pump from the cylinder, when the pressure is to be released, a small check valve controlled by a lever or a screw is usually provided. Machines operating hydraulically are not

satisfactory for large loads, because the leakage from the cylinder is likely to be excessive.

When a specimen is to be tested in **tension** its upper end is fastened into the wedges or "jaws" in the upper "head" and its lower end is similarly gripped in the lower or movable cross-head. For tests in **compression** the specimen is placed between the movable cross-head and the table. **Transverse loads** can be applied to long wooden or metal beams with the machine shown in **Fig. 256,** by placing the beam between the supports or abutments **UU′**, and applying the load by means of the movable cross-head **B.** Usually a special fitting with a blunt but "definable" edge to localize the load is inserted into the cross-head **B** for such tests.

Extensometers. Some of the physical properties of materials are determined by the rate of deformation of the specimen as the stress is applied. To measure the deformation some very accurate instruments have been devised, one of which is shown in **Fig. 257.** It consists essentially of a pair of clamps **CC′**, fitted with sharp-pointed thumb-screws for attaching them to the specimen **SS.** Two rods **B** and **B′**, fitted to the upper clamp on opposite sides of the specimen, are provided at their lower ends with adjustable points to be screwed up or down by means of the small milled wheels **W** and **W′**. Opposite these rods and fastened to the lower clamp are two micrometer screws, usually graduated to **ten-thousandths of an inch,** for measuring the elongation of the specimen. Electrical connections are made, as shown, with a battery and a bell. As the specimen stretches out the contact points at **P** and **P′** are moved apart and the distance the **micrometer screws** are raised measures the elongation. With the help of the bell

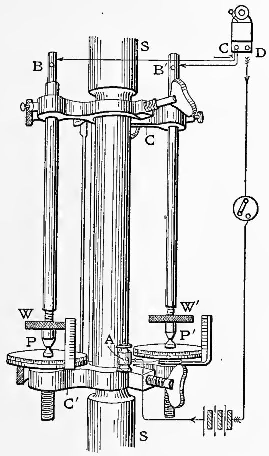


FIG. 257.—Extensometer.

it is possible to make, for all observations, uniformly light contacts.

Deflectometer. A very simple device for measuring the deflection of beams is shown in **Fig. 258,** consisting of a plate **P** supported upon a steel bar attached to the end supports **UU′.** Deflections can be measured with this apparatus with

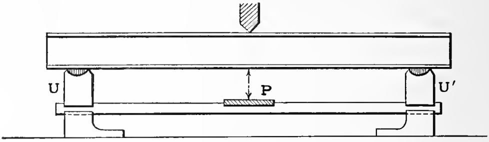


FIG. 258.—Simple Device for Measuring the Deflection of Beams.

the aid of ordinary "inside" calipers, micrometer calipers, or with a special deflectometer. This instrument, illustrated in **Fig. 259,** is often used to measure the deflection of wooden and metal beams subjected to transverse stress. It can also be used with success to measure the contraction of short specimens in compression.

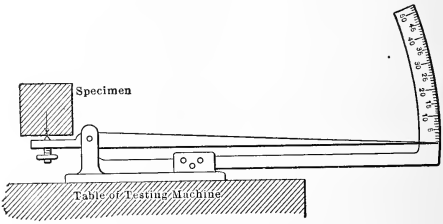


FIG. 259.—Deflectometer.

Physical Properties of Materials Defined. The **elastic limit** is a more or less definite value of the unit stress beyond which, as the stress is increased, the increase in deformation is greater relatively than the increase in stress; and further, at this point, the deformations produced will not disappear entirely when the stress is removed. **Permanent set** or **" set "** is used to represent the lasting deformations produced by stresses greater than the elastic limit.

Modulus of Elasticity is a term used to express the ratio of the unit stress to the deformation per unit of length [1] accompanying that stress, within the elastic limit. For example, if **f** is the stress in pounds per square inch within the elastic limit and **s** is the accompanying **deformation per inch of length** in inches, then the modulus of elasticity, **in pounds per square inch, is**

$$\mathbf{E} = \mathbf{f}/\mathbf{s} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \qquad (122)$$

The total stress under which a body fails is called its ultimate strength; and the corresponding unit stress is called the ultimate unit strength, or, for short, simply the **ultimate strength.** The ratio of the total elongation of a body to its original length is called the **percentage of elongation.** It is obviously the same as the term unit deformation. For calculations of percentage of elongation, measurements are taken, according to convention, between two gage marks usually 8 inches apart. This percentage of elongation is a measure of the **ductility** of the material tested. The **ratio** of the smallest area after rupture to the original area is called the **"reduction of cross-section."**

Resilience, often called the modulus of resilience, is a term used to represent the potential energy stored in a body; or, from another viewpoint, it is the amount of work that can be done by the body when relieved from a state of stress. More specifically however, it is taken to mean in practice the **work, in foot-pounds,** done on a cubic inch of a material in stressing it to the **elastic limit.** For any value of the load, the resilience is equal to the product of half the stress in pounds per square inch at the elastic limit times the distortion (usually the elongation) of the test-piece in feet per inch of length up to the elastic limit, the latter term being the space passed through. Values of resilience calculated as thus defined may be **checked** by comparing with the square of the unit stress at the elastic limit divided by 24 times the value of the modulus of elasticity (**E**).

Forms of Test-pieces for Tension Tests. Specimens for testing should be prepared with a great deal of care. Standard forms for **round and flat bars** [2] **for tension tests** are shown in **Figs.**

[1] This unit deformation is often called the unit elongation for slender test-pieces, and more generally the **strain**.

[2] See "Materials of Construction," J. B. Johnson.

260 and **261.** On such test-pieces marks one inch apart are usually made between the limits of the so-called gage length **AB,** which is generally 8 inches. A standard scale similar to the one in **Fig. 262** is of great assistance in marking a test-piece. At the left-hand end a percentage scale is shown from which the

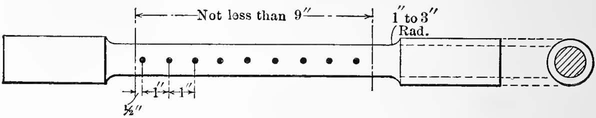


FIG. 260.—Standard Round Bar for Tension Tests.

percentage of elongation, in a length of 8 inches, can be read directly.

The relation between the **gage length AB,** and the diameter is for round sections, **l = 8d,** and for square sections **l = 9bt.** Machine work on specimens for testing should be done carefully,

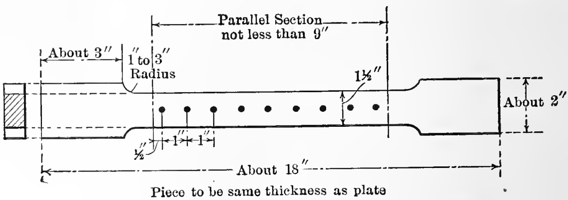


FIG. 261.—Standard Flat Bar for Tension Tests.

so that the material is not torn or weakened in other ways. If there is any flaw, marked irregularity or other defect in the material, the test-piece should be rejected. After, however, a test-piece has been "necked" and broken as shown in **Fig. 263,** the elongation cannot be measured very accurately with such a scale. One method is to measure the elongation from the point of rupture toward each end. In materials in

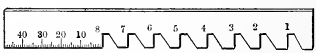


FIG. 262.—Scale for Marking Test-pieces.

which the "necking" effect is very marked the measured amount of elongation will vary according to the distance of the fracture

from the **gage marks A** and **B.** If the fracture is midway between these marks then nearly all the elongation will be between these marks; but if the fracture is near one of the gage marks then a great deal of the elongation will fall outside of the marks, so that the measured elongation is too small. To correct for these discrepancies mentioned, a so-called "equivalent elongation" is calculated by the following method:

Assume that the standard test-piece, **Fig. 260,** has been divided originally into 8 equal spaces between the gage marks **A** and **B,** and that the nearest number of spaces between the points of fracture (**Fig. 263**) and the nearer gage mark is 3; or in other words, that there are 3 spaces **on the shorter portion** of the specimen between the point of rupture and the gage mark **B.** Then the **total** length to compare with the original is to be measured on the broken test-piece from the point 5 to **B** on the right (corresponding to 3 spaces), plus from 4 to 5 on the right,

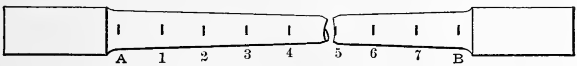


FIG. 263.—Test-piece (Round Bar) after Rupture.

plus the distance from **B** to 4 on the left. The sum of these lengths will be the "equivalent" total length after rupture. In general terms the method may be stated that if the length between gage marks has been divided into **x** equal spaces and **y** is the **nearest** number of these spaces on the shorter portion between the point of rupture and the gage mark, then mark two points **M** and **N** on the **longer** portion, which are **y** and **1/2x** spaces, respectively, from the fracture. Place the two portions of the specimen as closely together as possible and measure from the gage mark in the short portion to the mark **M.** This distance, added to double the distance from **M** to **N,** gives the required total length after rupture. In this way the elongation of the "standard" length (**x** spaces) will be obtained, as if the fracture had occurred midway between the gage marks.

Specifications are often made to require that the fracture shall be within the "middle third of the length."

Detailed Method for Tension Tests. A standard test-piece to be tested in **tension** should be without flaws or cracks, and furthermore the material should be monogeneous. Before

putting it into the testing machine it should be carefully measured. With a scriber or scratch the marks indicating one-inch divisions should be made with the "laying-off" gage. (**Fig. 262**). Very light punch marks may then be made at each of the division marks accurately along the axis of the bar.[1] At each of these punch marks the diameter of the cross-section should be carefully measured with a micrometer caliper to thousandths of an inch. The outside punch marks, called the "gage marks," are often made a little heavier than the others, so that if an extensometer of the type illustrated in **Fig. 257** is used, one of the thumbscrews supporting the clamps at each end can be set accurately but lightly into these marks. In this position the extensometer will measure accurately the elongation of the specimen between the gage marks. The testing machine to be used should then be balanced by adjusting the counterpoise provided for this purpose. It should be observed also that the **"check nuts"** rest loosely on the table. As the load is increased, however, these nuts should be screwed down a little from time to time so that if the load is suddenly removed when the test-piece breaks, or should happen to slip out of the jaws or wedges holding it in place, the jar on the machine will be very much relieved.

After balancing the machine the test-piece should be placed carefully and vertically between the "jaws" or wedges in both the upper and lower heads, and the extensometer should be put in position if one is to be used. Start the machine at a low rate of speed till a load of about 1000 to 2000 pounds is indicated before taking any measurements of elongation. This first load is applied for the purpose of permitting the test-piece to assume a true central position, to allow for some slight slipping of the test-piece in the jaws before it becomes firmly gripped, and also to allow for possible irregularity in the adjustment of the extensometer or similar auxiliary apparatus. After this light load has been reached, during which time, doubtless for the kind of materials ordinarily tested, there will have been only a very small elongation, the load should be applied continuously and

[1] Punch marks can be made accurately along the axis of a round bar by putting them at the "scratch" marks made with a scriber an inch apart along the length of the test-piece on the reflection of a "beam of light" on the bright surface of the test-piece.

uniformly until the test-piece breaks, stopping only long enough, at the required intervals, to make the necessary observations of elongation and change of shape of the cross-section when "necking" begins. Increments of load are usually determined by taking one-tenth [1] the product obtained by multiplying the approximate estimated elastic limit of the material in pounds per square inch by the area of the cross-section in square inches.

Near the elastic limit it is often found desirable to take observations at intervals of 500 pounds up to the **yield point,** where suddenly the rate of elongation increases very rapidly. By taking these observations at very close intervals at the elastic limit, data are secured which make the curves to be plotted much more satisfactory.

In a **standard** test the stress, that is, the load, must never be **decreased** any appreciable amount when it is intended to apply again later an increased load. A stress once applied must be maintained or increased continuously till the end of the test. Extensometers or other apparatus of delicate construction used for measuring the elongation should be removed from the test-piece before the specimen is broken. It is customary to take off such apparatuses just after the elastic limit is reached; although, as a rule, they can be left in place relatively longer for materials that are ductile than for those that are hard and brittle. Micrometers used on extensometers provided with electrical connections are to be set for readings on one side at a time, advancing the screw device until the bell rings, indicating that the contact has been made. Then turn it back **just enough to stop the ringing of the bell,** and advance the screw on the other side till the bell rings again. After also turning back the micrometer screws **just enough to stop the ringing,** observations should be taken on both micrometers. Considerable time can be saved if, while these observations are being taken, the attendant having charge of the machine is meanwhile slowly advancing the load. The scale beam should be kept "floating" at all times during a test. Never run the balancing poise out on the scale beam beyond the point necessary to balance the

[1] In practice the **increments** of load in **commercial** tests are often one-half or one-third of the load at the elastic limit. For laboratory investigations it is not unusual to make the increments as small as one-twentieth of this load.

beam. If the scale beam is carefully kept " floating " a point will be observed at from 50 to 75 per cent of the maximum load where the scale beam will fall, indicating **apparently** that it has been advanced too far. This point is called the **yield point.** It is defined as the stress at which the rate of elongation suddenly and rapidly increases.

Beyond the elastic limit, when the extensometer has been removed, the rate of elongation can be measured with considerable accuracy by means of a large machinist's dividers, with the points set accurately on the **" gage marks "** at the ends of the standard length.

The load when rupture occurs is not usually the **maximum.** When, therefore, considerable " necking " effect is observed, the poise on the scale beam should be watched closely and when the maximum load has been reached, indicated by the falling of the scale beam, this weight should be quickly observed and recorded and then the poise should be brought back to follow the decreasing load. It will be observed that **this part of the work is very interesting if it is done carefully.**

After the test-piece has been broken, stop the machine, remove the test-piece, clean and return the jaws or wedges to their proper places, and **leave the machine in good order.**

The broken ends of the test-piece should be joined carefully and the length between each of the marks which were originally one inch apart should be measured. Check the sum of these lengths with the overall length between the gage marks. With a micrometer, or preferably a vernier caliper, measure as accurately as possible the diameter of the smallest area at the fracture. The fracture should be carefully examined to observe whether it is fibrous, granular or crystalline; whether coarse, fine or "silky," whether cup-shaped, half-cup, or irregular in shape.

Curves and Calculations. Plot a curve with elongation per inch of length (" strain ") as abscissas and stress in pounds per square inch as ordinates. This is the familiar " stress-strain " diagram.

Determine from the data obtained the modulus of elasticity (**E**),[1] the elastic limit, the maximum stress, the ultimate stress,

[1] The modulus of elasticity can be determined also from the " stress-strain " diagram by calculating the **value of the tangent** for the angle between a line drawn through the origin parallel to the straight part of

per cent elongation in 8 inches, percentage elongation in 2 inches at the fracture, and percentage reduction in area at the fracture.

Plot a curve of elongation per inch, using for abscissas [1] the original length in inches and for ordinates the elongations measured for each inch between the gage marks.

REPORT OF TENSION TESTS

1. Date, and names of observers....................................
2. Material to be tested—specification..............................
3. Makers and brand of material....................................
4. Length between principal punch marks, before test................
5. Length between principal punch marks, after test.................
6. Average width of test-piece, before test (6 readings)..............
7. Average width of test-piece, after test (6 readings)................
8. Average thickness of test-piece, before test (9 readings)...........
9. Average thickness of test-piece after test (6 readings).............

   Readings to be taken, for observations 10 and 11 for increments of 2000 lbs. on scale beam until an elongation of 0.04 in. is obtained between readings and then by increments of 1000 lbs. until elongation of 0.03 in. is obtained between readings, and finally by 500 lbs. until piece nears point of maximum strength, when readings should be taken as frequently as possible, keeping beam balanced all the time.

10. Total pull in pounds as shown by the machine...................
11. Corresponding total elongation in piece in inches.................
12. Yield point, pounds total (elastic limit)........................
13. Maximum load, pounds total....................................
14. Breaking load, pounds total....................................
15. Character of fracture, description and sketch......................

    The following results are to be computed:

16. Original cross-section, sq.in....................................
17. Final cross-section, sq.in.......................................
18. Reduction in area in sq.in., and in per cent of original area..........
19. Final elongation, total and per cent.............................
20. Stress (pounds per square inch of original area) at elastic limit, maximum load and breaking load.

Tests in Compression. When a specimen of which the length is less than five times the smallest dimension is subjected to a load producing compression, it fails usually by **crushing.** Longer specimens fail usually by bending toward the side of least resist-

"stress-strain" curve, reading the scales of co-ordinates, of course, in the units of stress and elongation marked on the diagram. It is obvious that the value of the tangent of the angle in this case is the unit stress divided by the unit elongation, which is, by definition, the modulus of elasticity.

[1] If on the curve sheet the "inch marks" on the test-piece are indicated by equal divisions on the scale of abscissas, then the points showing elongation for each inch should be plotted midway between the division lines indicating the position of the "inch marks."

ance. Two general classes of materials are frequently tested in compression: (1) **Brittle** materials, like brick, stone, wood, cement, cast iron, etc., which fail usually by **shearing,** and (2) **plastic** materials, like soft steel, wrought iron, copper, etc., which fail usually by a "flowing" of the metal. Because of the difficulty in measuring the deformations of short specimens of the plastic materials, and because the elastic limit in tension is invariably practically the same as in compression, these plastic materials are not often subjected to compressive loads. Methods to be explained here apply, therefore, particularly to materials like wood, brick, stone, and cast iron.

Detailed Method for Compression Tests of Short Test-pieces. Specimens of stone, cement, wood, or brick, of which the length is less than five times the smallest dimension, are **usually** provided in forms approximately cubes, although brick and wood are as often tested in the form of parallelopipeds similar to ordinary commercial bricks. Bearing surfaces of specimens of **stone, brick,** and **cement** should be made as nearly flat and parallel as possible, and should then be covered with a thin layer of plaster of Paris. Sized paper in thin sheets should be placed on the bearing surfaces between the specimen and the plaster to prevent the absorption of water from the latter. In order to have the plaster set in a true surface the specimen is placed between the "heads" of the testing machine for about ten minutes after the movable cross-head (**B, Fig. 256**) has been lowered to press lightly on the plaster. For tests in compression the test-piece is placed on the table **T** and the load is applied by lowering the movable cross-head **B.**

Dimensions of test-pieces must be carefully measured and recorded before they are put into the testing machine; and, if any of them require the application of plaster of Paris, then the measurements must be made before the plaster is put on. After balancing the testing machine by means of the counterweight **with the test-piece on the table,** apply the load continuously until the specimen is fractured; or in the case of plastic materials till the deformation is quite noticeable. In general conduct the test in the same way as for tension,[1] except that the specimen

[1] Measurements of the amount of compression (shortening) of the test-piece cannot be made directly, but must be made between points on the heads of the testing machine. If there is likely to be much

is compressed instead of being stretched. If the material tested is cast iron or even hard stone or brick, precautions must be taken to protect persons near the machine from flying fragments. If the specimen begins to spall or flake off before it breaks down, the load corresponding as well as similar information should be recorded and placed in the tabulated report under " Remarks." Usually the specimen breaks down suddenly and the interior cone or pyramid in stone, brick, cement, and cast iron will be plainly seen if the load has been " fairly " applied. In the case of tests of wood, this phenomenon will not be observed, but the lines of cleavage will usually show clearly a constant angle of shearing.

Detailed Method of Compression Tests of Long Pieces (Columns). When the length of a specimen to be tested in compression is greater than, at the most, ten times its least dimension, it fails invariably by bending toward the side of least resistance. The condition of the ends of such test-pieces should be as nearly as possible either fixed or perfectly free to turn. Either condition is, however, difficult to obtain. For test-pieces from 15 to 20 inches long usually an extensometer may be connected up to read the compression or shortening of the test-piece, if it is desired.[1] The observations will be taken in the same general way as for tension tests except that now the micrometer screws on the extensometer will approach each other, so that these screws must be turned back after taking a measurement by an amount greater than the compression that will be produced by the next increment of load.

yielding of the parts of the machine, the moving head should be lowered till its steel " compression plate " presses on the corresponding steel block on the table or lower platform with a force of about 1000 pounds. Now measure with micrometers the distance between the points on the two heads used for compression measurements, first with the load of 1000 pounds and then with additional increments of 1000 pounds up to considerably above the breaking load of the material to be tested. From these data a correction curve should be plotted with which to correct the deflections observed when the specimen is tested.

When blocks of wood are to be tested in compression, readings of the micrometers should not be taken till a pressure of 500 to 1000 pounds **per square inch** has been applied. This load will be required to crush the rough fibers.

[1] The lateral deflection along the neutral plane is sometimes determined by stretching a fine wire along the length of the specimen parallel to the neutral axis.

Report on Compression Tests

1. Date and names of observers ....
2. Kind of material ....
3. Average thickness of test-piece (4 readings), inches ....
4. Average width of test-piece (4 readings), inches ....

Machine is to be started and kept running continuously until fracture takes place, the beam being kept balanced carefully all the time. Readings to be taken, and calculations made therefrom as follows: In making these tests, wood and brick will be used and two pieces of each kind are to be tested, with each kind of stress.

Kind of Stress	Tension.					
Kind of Wood	White Pine.		Yellow Pine.		Bricks.	
Number of Piece	1	2	1	2	1	2
Scale reading in pounds at time of fracture						
Character of fracture, sketch.						
Cross-section from items 3 and 4, square inches						
Breaking stress lbs. per square inch for each piece						
Average breaking stress for each kind of wood						
Modulus of elasticity, lbs. per square inches						

Sketches, Curves, and Calculations. Sketch the character of the fracture for each specimen tested, indicating, **for wood,** the direction of the grain. Previously, the original shape of the specimen should have been sketched and dimensioned.

Calculate the **maximum** unit stress.

If the material was suitable for the measurement of compression, plot "stress-strain" diagrams, and calculate the modulus of elasticity.

Transverse Bending Tests. The most common test by transverse or cross-bending is that of a beam usually of either wood or steel, of which the coefficient of elasticity and the "elastic curve" are desired. Deflections of such beams give the data needed. Such tests may be made with a testing machine like the one shown in **Fig. 256,** which is provided with supporting abutments marked in the figure **UU′,** and by inserting into the movable head the attachment for applying the load

along a line across the beam rather than on a comparatively large area as in the tests already described when the load was applied directly by the flat surface of the cross-head **B.** Special transverse testing machines are, however, sometimes available. A machine of this kind is illustrated in **Fig. 264.**

In the case of a **wooden** beam to be tested by loading at the middle, a fine steel wire should be stretched between two pins

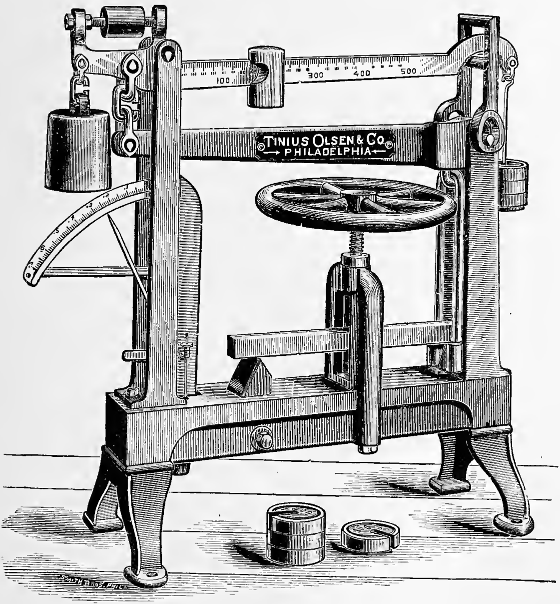


FIG. 264.—Machine for Transverse Tests.

located as accurately as possible above the points of support and on the line of intersection of the neutral plane with the side of the beam. The wire should be fastened to one of these pins and allowed to hang over the other, being kept taut by means of a weight attached to the free end, **Fig. 265.** A steel scale, preferably highly polished so that it will show the image of the wire, should be attached in any suitable way to the side of the

beam, so that the edge along which the scale readings to be observed are marked will be exactly half-way between the two supports. The beam must be protected from indentation by the knife-edges by small bearing plates. The load should be applied centrally in increments to give approximately $\frac{1}{50}$ inch deflections to the elastic limit, and beyond to give deflections of approximately $\frac{1}{25}$ inch. If it can be done successively, the deflections should be read without stopping the test; unless, of course, the permanent set is to be determined, when after each increment, the beam must be released from its load.

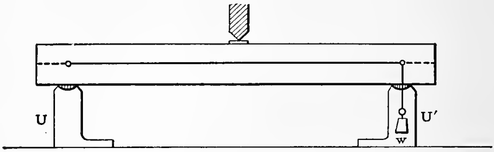


FIG. 265.—Device for Measuring the Deflection of a Wooden Beam.

Curves and Calculations. Plot a curve taking the load applied in pounds for abscissas and deflections in inches for ordinates.

Sketch the character of the fracture.

Calculate the modulus of elasticity,[1] the modulus of rupture, and the stress in the outer fiber at the elastic limit from the curve.

[1] The modulus of elasticity is calculated by the formula

$$E=\frac{w_e l^3}{48dI}. \qquad (123)$$

The modulus of rupture from

$$f_u=\frac{w_u lc}{4I}, \qquad (124)$$

and the **stress in the outer fiber at the elastic limit** by

$$f_e=\frac{w_e lc}{4I}, \qquad (125)$$

when w_e = load at the elastic limit in pounds per square inch;
w_u = load at the point of rupture in pounds per square inch;
l = length of beam (span) in inches;
d = deflection in inches;
c = distance from the neutral axis to the outer fiber in inches;
I = moment of inertia, inches.

Torsion Tests are made to determine the strength of a material to resist twisting forces. A typical machine for such tests is illustrated in **Fig. 266.** It consists in its essential parts of the frame **FF′,** the "jaw" heads **A** and **B** for gripping the rod **R** to be tested, and the system of weighing levers on which the poise **P** is balanced. The load is applied to the rod **R** by power through the gears shown connected to the head **B.**

The power is applied by means of the pulley and gears shown at the right-hand side, or may be applied by hand power by turning hand wheels. With hand power usually more satisfactory results can be obtained than with power applied

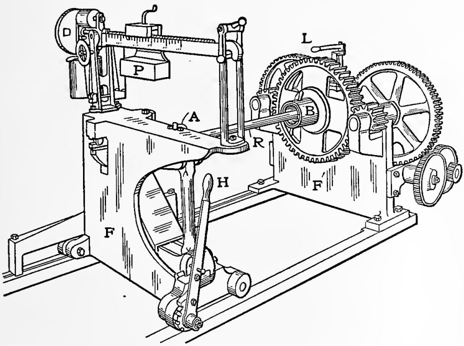


Fig. 266.—Riehlé Torsion Testing Machine.

mechanically, because the rate of twisting can be more closely regulated. The amount of twist or the angular deformation is indicated by index-arms connected to opposite ends of the test-piece.

An autographic torsion testing machine operated by hand power by means of the crank is sometimes used. The movement of the crank tends to rotate the test-piece which at the opposite end of the machine is fastened to a pendulum carrying a heavy bob. The resistance of the pendulum and its weight measure the power applied, which is equal to the length of the lever arm times the sine of the angle of inclination times the constant weight of the bob.

Tests are made usually by increasing the twisting moment by increments of about 200 **inch-pounds**,[1] measuring for each increment the torsion angle.

Curves and Calculations. Plot a curve, using torsion angle for abscissas and twisting moment for ordinates. Calculate from this curve the unit stresses[2] (shearing) at the elastic limit, at the point of rupture, and also the maximum value of stress. Determine the torsion angle at the elastic limit and at the point of rupture, the helix angles,[3] and the modulus of elasticity **for torsion.**[4]

Impact Tests. Materials are tested by impact, usually by striking a test-piece with a weight allowed to fall upon it. Metals used in the manufacture of machinery and in railroad construction where it is likely to be subjected to shocks and blows are in many cases tested to determine the effect of the impact due to a blow.

Some testing machines for such tests are made like a pile-driver with the weight dropping vertically from a sort of gallows upon the test-piece. A more common form, however, of such machines consists of a pendulum provided with a heavy bob

[1] If tests are made of large sections of high-grade material, like, for example, a shaft of nickel steel for a students' class, it is expensive to break many specimens, so that for this reason the twisting moment producing a maximum stress just inside the elastic limit is computed before making the test, and this value is not to be exceeded.

[2] The unit stress is calculated with the formula:

$$\mathbf{M}/\mathbf{f}_s = \mathbf{I}_p/\mathbf{c}, \quad \text{or} \quad \mathbf{f}_s = \mathbf{Mc}/\mathbf{I}_p, \quad . \quad . \quad . \quad . \quad (126)$$

where **M** is the torsional moment in inch-pounds, **c** is distance in inches from the neutral axis to the extreme fiber, and $\mathbf{I}_p$ is the polar movement of inertia. When $\mathbf{c}=\mathbf{r}$ (the radius) as in the case of a **cylindrical** test-piece, $\mathbf{I}_p = \frac{1}{2}\pi\mathbf{r}^4$.

[3] Torsion produces a peculiar arrangement of the outer fibers in the form of helices, as observed in broken test-pieces. Each one of these fibers makes an angle with its original position equal to its angular distortion α. Any particle on the surface is also moved through an angle β, having its vertex in the axis and in a plane perpendicular to the axis. Now if we neglect the functions of small angles, we can write approximately $\mathbf{l}\alpha = \mathbf{r}\beta$, where **l** is the effective length of the test-piece and **r** is the radius. The helix-angle $\alpha = \mathbf{r}\beta/\mathbf{l}$.

[4] The modulus of elasticity in torsion ("modulus of rigidity"),

$$\mathbf{E}_s = \mathbf{f}_s \div \alpha,$$

as above, then

$$\mathbf{E}_s = \frac{\mathbf{f}_s \mathbf{l}}{\mathbf{r}\beta}. \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (127)$$

intended for delivering a blow on the middle of a test-piece in the shape of a bar, preferably of a rectangular section, held on two knife-edge supports attached to a heavy bedplate. Such machines are particularly designed for comparative tests of cast iron. They are provided with an arc concentric with the movement of the bob of the pendulum, graduated to read the vertical fall of the bob in feet. A tripping device is attached to the side of the graduated arc for permitting the bob to be supported and then dropped from any height within the limits of the machine. Since the deflection is very small, a device is usually supplied for magnifying it, and by means of a pencil-point traveling over a chart an autographic record is made of the deflections for each blow delivered by the bob. With such instruments the rebound of the test-piece and its permanent set must be carefully excluded from the measured deflection. One way to do this is to draw a "zero line" with the test-piece in place but before a blow is struck. Deflections and permanent set will then be measured on one side of this line and "rebounds" on the other.

To determine the center load to be applied that will be equivalent to the impact, the following symbols are used:

Let $\mathbf{w}_1$ = weight of the bob in pounds;
$\mathbf{h}$ = the vertical distance it falls, in feet;
$\mathbf{w}_2$ = the equivalent maximum center load, in pounds, and
$\mathbf{d}$ = the deflection in feet, then

$$\mathbf{w}_1\mathbf{h} = \tfrac{1}{2}\mathbf{w}_2\mathbf{d},$$

$$\mathbf{w}_2 = \frac{2\mathbf{w}_1\mathbf{h}}{\mathbf{d}}. \qquad (128)$$

With this value of $\mathbf{w}_2$ the usual properties of the material may be calculated by formulas (123), (124) and (125), page 392.

Cement Tests. Cements are tested usually for tensile and crushing strength, for fineness, and for the time required for "setting." Tests for crushing strength (compression) are usually made by crushing cubical blocks in a testing machine designed for general tension and compression tests (see page 378). For tensile tests, however, special machines, designed particularly for testing cement, are generally used. Because of the nature of the material it is absolutely necessary that the power is

applied in the line of the axis of the test-piece and also with steadiness and in increments as uniform as possible. There is a standard size and shape for test-pieces of cement, and they must be made in a certain prescribed way in order that different tests may be compared. The standard briquette for testing

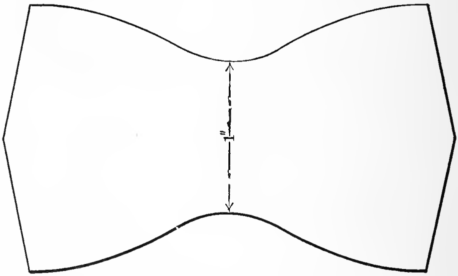


FIG. 267.—Standard Specimen for Cement.

(one square inch section) is shown in **Fig. 267,** and **Fig. 268,** shows moulds suitable for making the test-pieces or **briquettes,** as they are called.

Cement-testing machines are invariably provided with moulds for making standard briquettes. These moulds are divided

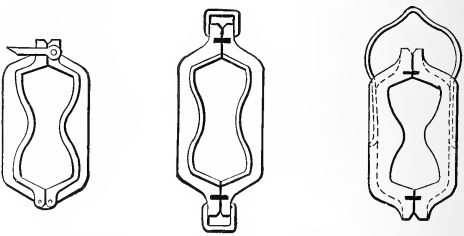


FIG. 268.—Cement Moulds and Briquettes.

along a longitudinal center line into two halves which, when placed together, fit closely and are held in place by means of pins, or dowels, preventing endwise movement, and by clamps pressing together the two parts of the mould. The strength

of the briquettes is affected by the time allowed for hardening, the amount of water used, and by the method of mixing the cement. (See page 400.)

Power is applied in the **automatic** cement-testing machine shown in **Fig. 269,** by shot dropped from a cylindrical hopper into a pail supported on a scales. The briquette of cement

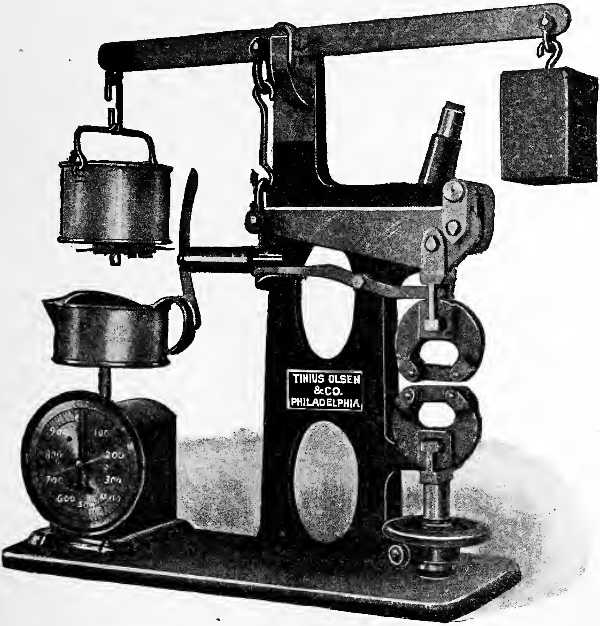


FIG. 269.—Automatic Cement Testing Machine.

being tested is held between two shackles or " holders " connected to a hand wheel used to regulate the distance between the shackles. When a briquette breaks the scale beam drops and closes automatically a valve, stopping the delivery of shot into the pail. The operation of the machine may be described briefly as follows: Hang the hopper on the hook as shown and put enough shot into it to balance the counterpoise. Now

put the briquette into the shackles and adjust the hand wheel so that the scale beam will rise nearly to the stop. When the valve is opened shot will begin to fall into the pail. The delivery of the shot into the pail should be slow. When the briquette has broken, the scale beam has dropped and the valve

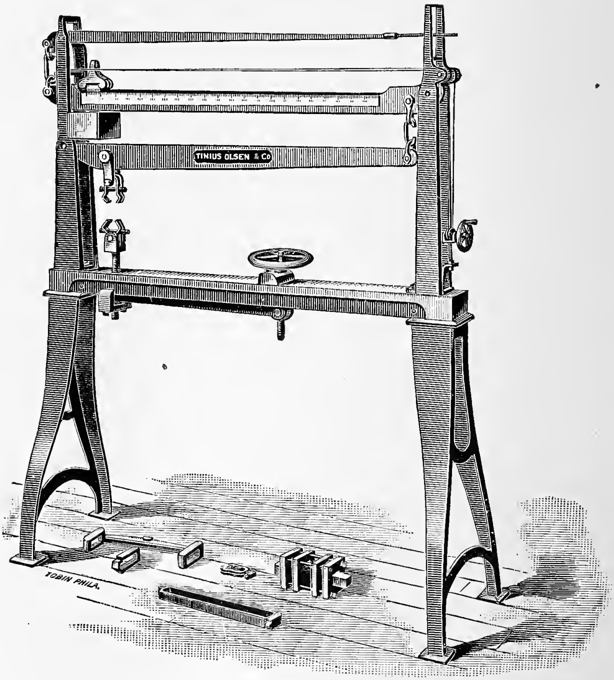


FIG. 270.—Hand-operated Cement Testing Machine.

has been closed. The weight of shot collected in the pail shows the number of pounds required to break the briquette.

A non-automatic type of cement-testing machine is illustrated in **Fig. 270.** In this machine the power is applied by moving a hand wheel operating by means of a screw the system of levers transmitting the load to the briquette in the

shackles. The tension produced by this load may obviously be balanced by weights applied to the scale beam above. In order to secure a very uniform and slow movement of the poise it is carried along the scale beam by a cord moved by a small crank. In applying the load the hand wheel should be moved as slowly and uniformly as possible to avoid a jerking motion. In the figure one of the standard briquette moulds, and a tray used for immersing the briquettes in water after they have set are shown. **Fig. 271** shows the proper position of the briquette in the supporting shackles.

Test of Cement for Fineness is made by determining the amount by weight of a given sample that will **not** pass through sieves with meshes of a standard size. The American Society of Civil Engineers recommends the use of sieves of 2500, 5476 and 10,000 meshes per square inch. Sieves with approximately these meshings are known as Nos. 50, 80, and 100. A weighed sample of cement is first passed through No. 50 sieve and the weight of that remaining in the sieve is recorded. That passing through is then put into the next sieve (No. 80) and the residue in this sieve is likewise weighed, while that passing through goes to the finest sieve (No. 100). Results of this test for fineness are expressed by the percentages that the various residues remaining in the sieves are of the original weight of the sample.

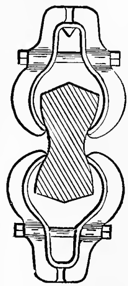


Fig. 271.

Unless cement is ground as fine as flour it has very little "binding power." The coarse particles are nearly as inert for "cementing" as sand.

Test of Cement for Time of Setting is made by mixing on a slab of glass circular pats about 3 inches in diameter and one-half inch thick. Then when a blunt needle (one-twelfth inch in diameter as the point) and loaded with a weight of one-quarter pound "ceases to penetrate the entire mass setting is said to have begun." Similarly when a needle one twenty-fourth inch in diameter loaded with a weight of one pound **ceases to penetrate at all,** setting is said to be ended.

Making Briquettes. Neat Cement. Before filling moulds with cement they should be carefully cleaned and rubbed on the

inside surface with a rag saturated with kerosene. Then clamp the parts together, placing the moulds preferably on a large slab of glass, or a similar material providing a plane surface which will not absorb moisture from the briquettes. A quantity of cement, enough to fill several moulds, should now be mixed with water (usually not as much as 25 per cent by weight) to make a rather stiff but very plastic and homogeneous mixture. It should be "worked" for at least three minutes. Put this mixture into moulds, pressing it down firmly, especially around the sides, preferably with the thumbs, in order that the pressure exerted will not be excessive. Level the surfaces of the briquettes with a trowel, being careful that in this operation too much is not taken from the middle so as to make the briquettes of unequal thickness. Usually cement briquettes are not tested in tension until several days after they have been made, and in that case the moulds with the briquettes in them should be allowed to remain on a slab which will not absorb moisture from them for twenty-four hours, in an atmosphere which is not very dry. After that time the briquettes should be carefully removed from the moulds, put into a suitable tray, like the one shown in **Fig. 270,** and immersed in a tank containing water.[1] Near one of the ends, each briquette should be marked with significant numbers or letters, so that the maker and the date of making will not be confused. The cement must not, of course, be permitted to begin to set before it is put into the moulds.

Mortar Briquettes for testing are made usually of five, two, or three parts of sand to one part of cement. The sand to be used is to be approximately of such fineness that it will all pass through a No. 20 sieve and is all held on a No. 30 sieve. Mortar briquettes for standard tests are made of pure crushed quartz of the kind used in the manufacture of sandpaper. For mortar briquettes less water will be needed than for those of neat ("pure") cement. The mortar briquettes should be worked with the trowel for at least four minutes. Moulds should be well filled with the mortar, which should then be pressed down to a flat surface with the trowel. The briquettes are to be set aside and later immersed in water as specified for those of neat cement.

[1] The water in this tank should be changed once in seven days, and should be kept at normal "room" temperatures.

For **laboratory** tests, usually after seven days, or else after twenty-eight days, the briquettes are taken from the water and placed in a cement-testing machine for testing in tension. Apply the load at as uniform a rate as possible, without jerks, and if a non-automatic machine is used the scale beam must be kept "**floating**" all the time, so that when the briquette breaks the correct load will be indicated by the position of the poise on the scale beam. In some laboratories it is recommended that pieces of thin rubber bands be inserted between the edges of the shackles or "holders" and the briquette, as in this way it is claimed there is greater certainty of having it break at the middle, that is, in the smallest section where the area is supposedly exactly one square inch.

Data regarding tests of neat cement and mortar briquettes should be tabulated in a form similar to the following:

FORM FOR CEMENT TESTS

1. Date, and names of observers.
2. Name and kind of cement.
3. Makers and location of plant.
4. Distinguishing mark on briquettes.
5. Date of mixing and time.
6. Temperature of room at time of mixing, deg. F.
7. Temperature of water at time of mixing, deg. F.
8. Conditions of setting: as to time briquettes were left in dampened air before immersion, etc.
9. Activity of the cement or time of initial and final setting.[1]
10. Fineness of grinding.

	Kind of Briquette.	Neat.									Sand.								
12 13 14	Composition of Briquettes.	Per Cent of Cement....... Per Cent of Water........									Per Cent of Cement........ Per Cent of Water......... Per Cent of Sand..........								
	No. of Briquettes.	7 Day.				28 Day.				Remarks	7 Day.				28 Day,				Remarks
		1	2	3	4	5	6	7	8		1	2	3	4	5	6	7	8	
15	Time of test.																		
16	Breaking Strength, lbs.																		
17	Appearance of fracture—give sketch of each here.																		
18	Temp. of Room at time of test. deg.F.																		

In reporting the results of these experiments it is important that the effect of different percentages of water, sand, etc., and time of immersion, be fully discussed.

[1] The time between the end mixing and the successful resistance to penetration of the needle $\frac{1}{12}$ inch in diameter with $\frac{1}{4}$ pound weight is called the "time of initial setting." Similarly, the time elapsing between the end of mixing and the resistance to penetration of the needle $\frac{1}{24}$ inch in diameter with 1 pound weight is called the "time of final setting."

APPENDIX

Two sets of tables which the author has found useful for "rough and ready" calculations are given on the following pages. Table I is a short table of the more important properties of saturated steam. This table has been taken with permission from Allen and Bursley's *Heat Engines*.

Table II gives for various common substances the specific gravity, density (weight per cubic foot), specific heat and coefficients of expansion per degree Fahrenheit, both linear and volumetric. Coefficient of volumetric expansion is three times the linear coefficient of expansion.

TABLE I

Properties of Saturated Steam

ENGLISH UNITS

Abs. Pressure Pounds per Sq. in.	Temperature Degrees F.	Heat of the Liquid	Latent Heat of Evaporation	Total Heat of Steam	Specific Volume Cu. Ft. per Pound	Density Pounds per Cu. Ft.	Abs. Pressure Pounds per Sq. In.
p	t	h	L	H	v	$\frac{1}{v}$	p
.0886	32	0	1072.6	1072.6	3301.0	.000303	.0886
.2562	60	28.1	1057.4	1085.5	1207.5	.000828	.2562
.5056	80	48.1	1046.6	1094.7	635.4	.001573	.5056
1	101.8	69.8	1034.6	1104.4	333.00	.00300	1
2	126.1	94.1	1021.4	1115.5	173.30	.00577	2
3	141.5	109.5	1012.3	1121.8	118.50	.00845	3
4	153.0	120.9	1005.6	1126.5	90.50	.01106	4
5	162.3	130.2	1000.2	1130.4	73.33	.01364	5
6	170.1	138.0	995.7	1133.7	61.89	.01616	6
7	176.8	144.8	991.6	1136.4	53.58	.01867	7
8	182.9	150.8	988.0	1138.8	47.27	.02115	8

Properties of Saturated Steam — *Continued*

ENGLISH UNITS

Abs. Pressure Pounds per Sq. In.	Temperature Degrees F.	Heat of the Liquid	Latent Heat of Evaporation	Total Heat of Steam	Specific Volume Cu. Ft. per Pound	Density Pounds per Cu. Ft.	Abs. Pressure Pounds per Sq. In.
p	t	h	L	H	v	$\frac{1}{v}$	p
9	188.3	156.3	984.8	1141.1	42.36	.02361	9
10	193.2	161.2	981.7	1142.9	38.38	.02606	10
11	197.7	165.8	978.9	1144.7	35.10	.02849	11
12	202.0	170.0	976.3	1146.3	32.38	.03089	12
13	205.9	173.9	973.9	1147.8	30.04	.03329	13
14	209.6	177.6	971.6	1149.2	28.02	.03568	14
14.7	212.0	180.1	970.0	1150.1	26.79	.03733	14.7
15	213.0	181.1	969.4	1150.5	26.27	.03806	15
16	216.3	184.5	967.3	1151.8	24.77	.04042	16
17	219.4	187.7	965.3	1153.0	23.38	.04277	17
18	222.4	190.6	963.4	1154.0	22.16	.04512	18
19	225.2	193.5	961.5	1155.0	21.07	.04746	19
20	228.0	196.2	959.7	1155.9	20.08	.04980	20
21	230.6	198.9	958.0	1156.9	19.18	.05213	21
22	233.1	201.4	956.4	1157.8	18.37	.05445	22
23	235.5	203.9	954.8	1158.7	17.62	.05676	23
24	237.8	206.2	953.2	1159.4	16.93	.05907	24
25	240.1	208.5	951.7	1160.2	16.30	.0614	25
26	242.2	210.7	950.3	1161.0	15.71	.0636	26
27	244.4	212.8	948.9	1161.7	15.18	.0659	27
28	246.4	214.9	947.5	1162.4	14.67	.0682	28
29	248.4	217.0	946.1	1163.1	14.19	.0705	29
30	250.3	218.9	944.8	1163.7	13.74	.0728	30
31	252.2	220.8	943.5	1164.3	13.32	.0751	31
32	254.1	222.7	942.2	1164.9	12.93	.0773	32
33	255.8	224.5	941.0	1165.5	12.57	.0795	33
34	257.6	226.3	939.8	1166.1	12.22	.0818	34
35	259.3	228.0	938.6	1166.6	11.89	.0841	35
36	261.0	229.7	937.4	1167.1	11.58	.0863	36
37	262.6	231.4	936.3	1167.7	11.29	.0886	37
38	264.2	233.0	935.2	1168.2	11.01	.0908	38
39	265.8	234.6	934.1	1168.7	10.74	.0931	39
40	267.3	236.2	933.0	1169.2	10.49	.0953	40
41	268.7	237.7	931.9	1169.6	10.25	.0976	41
42	270.2	239.2	930.9	1170.1	10.02	.0998	42
43	271.7	240.6	929.9	1170.5	9.80	.1020	43
44	273.1	242.1	928.9	1171.0	9.59	.1043	44
45	274.5	243.5	927.9	1171.4	9.39	.1065	45
46	275.8	244.9	926.9	1171.8	9.20	.1087	46

Properties of Saturated Steam — *Continued*

ENGLISH UNITS

Abs. Pressure Pounds per Sq. In.	Temperature Degrees F.	Heat of the Liquid	Latent Heat of Evaporation	Total Heat of Steam	Specific Volume Cu. Ft. per Pound	Density Pounds per Cu. Ft.	Abs. Pressure Pounds per Sq. In.
p	t	h	L	H	v	$\frac{1}{v}$	p
47	277.2	246.2	926.0	1172.2	9.02	.1109	47
48	278.5	247.6	925.0	1172.6	8.84	.1131	48
49	279.8	248.9	924.1	1173.0	8.67	.1153	49
50	281.0	250.2	923.2	1173.4	8.51	.1175	50
51	282.3	251.5	922.3	1173.8	8.35	.1197	51
52	283.5	252.8	921.4	1174.2	8.20	.1219	52
53	284.7	254.0	920.5	1174.5	8.05	.1241	53
54	285.9	255.2	919.6	1174.8	7.91	.1263	54
55	287.1	256.4	918.7	1175.1	7.78	.1285	55
56	288.2	257.6	917.9	1175.5	7.65	.1307	56
57	289.4	258.8	917.1	1175.9	7.52	.1329	57
58	290.5	259.9	916.2	1176.1	7.40	.1351	58
59	291.6	261.1	915.4	1176.5	7.28	.1373	59
60	292.7	262.2	914.6	1176.8	7.17	.1394	60
61	293.8	263.3	913.8	1177.1	7.06	.1416	61
62	294.9	264.4	913.0	1177.4	6.95	.1438	62
63	295.9	265.5	912.2	1177.7	6.85	.1460	63
64	297.0	266.5	911.5	1178.0	6.75	.1482	64
65	298.0	267.6	910.7	1178.3	6.65	.1503	65
66	299.0	268.6	910.0	1178.6	6.56	.1525	66
67	300.0	269.7	909.2	1178.9	6.47	.1547	67
68	301.0	270.7	908.4	1179.1	6.38	.1569	68
69	302.0	271.7	907.7	1179.4	6.29	.1591	69
70	302.9	272.7	906.9	1179.6	6.20	.1612	70
71	303.9	273.7	906.2	1179.9	6.12	.1634	71
72	304.8	274.6	905.5	1180.1	6.04	.1656	72
73	305.8	275.6	904.8	1180.4	5.96	.1678	73
74	306.7	276.6	904.1	1180.7	5.89	.1699	74
75	307.6	277.5	903.4	1180.9	5.81	.1721	75
76	308.5	278.5	902.7	1181.2	5.74	.1743	76
77	309.4	279.4	902.1	1181.5	5.67	.1764	77
78	310.3	280.3	901.4	1181.7	5.60	.1786	78
79	311.2	281.2	900.7	1181.9	5.54	.1808	79
80	312.0	282.1	900.1	1182.2	5.47	.1829	80
81	312.9	283.0	899.4	1182.4	5.41	.1851	81
82	313.8	283.8	898.8	1182.6	5.34	.1873	82
83	314.6	284.7	898.1	1182.8	5.28	.1894	83
84	315.4	285.6	897.5	1183.1	5.22	.1915	84
85	316.3	286.4	896.9	1183.3	5.16	.1937	85

PROPERTIES OF SATURATED STEAM — *Continued*

ENGLISH UNITS

Abs. Pressure Pounds per Sq. In.	Temperature Degrees F.	Heat of the Liquid	Latent Heat of Evaporation	Total Heat of Steam	Specific Volume Cu. Ft. per Pound	Density Pounds per Cu. Ft.	Abs. Pressure Pounds per Sq. In.
p	t	h	L	H	v	$\frac{1}{v}$	p
86	317.1	287.3	896.2	1183.5	5.10	.1959	86
87	317.9	288.1	895.6	1183.7	5.05	.1980	87
88	318.7	288.9	895.0	1183.9	5.00	.2002	88
89	319.5	289.8	894.3	1184.1	4.94	.2024	89
90	320.3	290.6	893.7	1184.3	4.89	.2045	90
91	321.1	291.4	893.1	1184.5	4.84	.2066	91
92	321.8	292.2	892.5	1184.7	4.79	.2088	92
93	322.6	293.0	891.9	1184.9	4.74	.2110	93
94	323.4	293.8	891.3	1185.1	4.69	.2131	94
95	324.1	294.5	890.7	1185.2	4.65	.2152	95
96	324.9	295.3	890.1	1185.4	4.60	.2173	96
97	325.6	296.1	889.5	1185.6	4.56	.2194	97
98	326.4	296.8	889.0	1185.8	4.51	.2215	98
99	327.1	297.6	888.4	1186.0	4.47	.2237	99
100	327.8	298.4	887.8	1186.2	4.430	.2257	100
101	328.6	299.1	887.2	1186.3	4.389	.2278	101
102	329.3	299.8	886.7	1186.5	4.349	.2299	102
103	330.0	300.6	886.1	1186.7	4.309	.2321	103
104	330.7	301.3	885.6	1186.9	4.270	.2342	104
105	331.4	302.0	885.0	1187.0	4.231	.2364	105
106	332.0	302.7	884.5	1187.2	4.193	.2385	106
107	332.7	303.4	883.9	1187.3	4.156	.2407	107
108	333.4	304.1	883.4	1187.5	4.119	.2428	108
109	334.1	304.8	882.8	1187.6	4.082	.2450	109
110	334.8	305.5	882.3	1187.8	4.047	.2472	110
111	335.4	306.2	881.8	1188.0	4.012	.2493	111
112	336.1	306.9	881.2	1188.1	3.977	.2514	112
113	336.8	307.6	880.7	1188.3	3.944	.2535	113
114	337.4	308.3	880.2	1188.5	3.911	.2557	114
114.7	337.9	308.8	879.8	1188.6	3.888	.2572	114.7
115	338.1	309.0	879.7	1188.7	3 878	.2578	115
116	338.7	309.6	879.2	1188.8	3.846	.2600	116
117	339.4	310.3	878.7	1189.0	3.815	.2621	117
118	340.0	311.0	878.2	1189.2	3.784	.2642	118
119	340.6	311.7	877.6	1189.3	3.754	.2663	119
120	341.3	312.3	877.1	1189.4	3.725	.2684	120
121	341.9	313.0	876.6	1189.6	3.696	.2706	121
122	342.5	313.6	876.1	1189.7	3.667	.2727	122
123	343.2	314.3	875.6	1189.9	3.638	.2749	123

Properties of Saturated Steam — *Continued*

ENGLISH UNITS

Abs. Pressure Pounds per Sq. In.	Temperature Degrees F.	Heat of the Liquid	Latent Heat of Evaporation	Total Heat of Steam	Specific Volume Cu. Ft. per Pound	Density Pounds per Cu. Ft.	Abs. Pressure Pounds per Sq. In.
p	t	h	L	H	v	$\frac{1}{v}$	p
124	343.8	314.9	875.1	1190.0	3.610	.2770	124
125	344.4	315.5	874.6	1190.1	3.582	.2792	125
126	345.0	316.2	874.1	1190.3	3.555	.2813	126
127	345.6	316.8	873.7	1190.5	3.529	.2834	127
128	346.2	317.4	873.2	1190.6	3.503	.2855	128
129	346.8	318.0	872.7	1190.7	3.477	.2876	129
130	347.4	318.6	872.2	1190.8	3.452	.2897	130
131	348.0	319.3	871.7	1191.0	3.427	.2918	131
132	348.5	319.9	871.2	1191.1	3.402	.2939	132
133	349.1	320.5	870.8	1191.3	3.378	.2960	133
134	349.7	321.0	870.4	1191.4	3.354	.2981	134
135	350.3	321.6	869.9	1191.5	3.331	.3002	135
136	350.8	322.2	869.4	1191.6	3.308	.3023	136
137	351.4	322.8	868.9	1191.7	3.285	.3044	137
138	352.0	323.4	868.4	1191.8	3.263	.3065	138
139	352.5	324.0	868.0	1192.0	3.241	.3086	139
140	353.1	324.5	867.6	1192.1	3.219	.3107	140
141	353.6	325.1	867.1	1192.2	3.198	.3128	141
142	354.2	325.7	866.6	1192.3	3.176	.3149	142
143	354.7	326.3	866.2	1192.5	3.155	.3170	143
144	355.3	326.8	865.8	1192.6	3.134	.3191	144
145	355.8	327.4	865.3	1192.7	3.113	.3212	145
146	356.3	327.9	864.9	1192.8	3.093	.3233	146
147	356.9	328.5	864.4	1192.9	3.073	.3254	147
148	357.4	329.0	864.0	1193.0	3.053	.3275	148
149	357.9	329.6	863.5	1193.1	3.033	.3297	149
150	358.5	330.1	863.1	1193.2	3.013	.3319	150
152	359.5	331.2	862.3	1193.5	2.975	.3361	152
154	360.5	332.3	861.4	1193.7	2.939	.3403	154
156	361.6	333.4	860.5	1193.9	2.903	.3445	156
158	362.6	334.4	859.7	1194.1	2.868	.3487	158
160	363.6	335.5	858.8	1194.3	2.834	.3529	160
162	364.6	336.6	858.0	1194.6	2.801	.3570	162
164	365.6	337.6	857.2	1194.8	2.768	.3613	164
166	366.5	338.6	856.4	1195.0	2.736	.3655	166
168	367.5	339.6	855.5	1195.1	2.705	.3697	168
170	368.5	340.6	854.7	1195.3	2.674	.3739	170
172	369.4	341.6	853.9	1195.5	2.644	.3782	172
174	370.4	342.5	853.1	1195.6	2.615	.3824	174
176	371.3	343.5	852.3	1195.8	2.587	.3865	176

Properties of Saturated Steam — *Concluded*

ENGLISH UNITS

Abs. Pressure Pounds per Sq. In.	Temperature Degrees F.	Heat of the Liquid	Latent Heat of Evaporation	Total Heat of Steam	Specific Volume Cu. Ft. per Pound	Density Pounds per Cu. Ft.	Abs. Pressure Pounds per Sq. In.
p	t	h	L	H	v	$\frac{1}{v}$	p
178	372.2	344.5	851.5	1196.0	2.560	.3907	178
180	373.1	345.4	850.8	1196.2	2.532	.3949	180
182	374.0	346.4	850.0	1196.4	2.506	.3990	182
184	374.9	347.4	849.3	1196.7	2.480	.4032	184
186	375.8	348.3	848.5	1196.8	2.455	.4074	186
188	376.7	349.2	847.7	1196.9	2.430	.4115	188
190	377.6	350.1	847.0	1197.1	2.406	.4157	190
192	378.5	351.0	846.2	1197.2	2.381	.4200	192
194	379.3	351.9	845.5	1197.4	2.358	.4242	194
196	380.2	352.8	844.8	1197.6	2.335	.4284	196
198	381.0	353.7	844.0	1197.7	2.312	.4326	198
200	381.9	354.6	843.3	1197.9	2.289	.4370	200
202	382.7	355.5	842.6	1198.1	2.268	.4411	202
204	383.5	356.4	841.9	1198.3	2.246	.4452	204
206	384.4	357.2	841.2	1198.4	2.226	.4493	206
208	385.2	358.1	840.5	1198.6	2.206	.4534	208
210	386.0	358.9	839.8	1198.7	2.186	.4575	210
212	386.8	359.8	839.1	1198.9	2.166	.4618	212
214	387.6	360.6	838.4	1199.0	2.147	.4660	214
216	388.4	361.4	837.7	1199.1	2.127	.4700	216
218	389.1	362.3	837.0	1199.3	2.108	.4744	218
220	389.9	363.1	836.4	1199.5	2.090	.4787	220
222	390.7	363.9	835.7	1199.6	2.072	.4829	222
224	391.5	364.7	835.0	1199.7	2.054	.4870	224
226	392.2	365.5	834.3	1199.8	2.037	.4910	226
228	393.0	366.3	833.7	1200.0	2.020	.4950	228
230	393.8	367.1	833.0	1200.1	2.003	.4992	230
232	394.5	367.9	832.3	1200.2	1.987	.503	232
234	395.2	368.6	831.7	1200.3	1.970	.507	234
236	396.0	369.4	831.0	1200.4	1.954	.511	236
238	396.7	370.2	830.4	1200.6	1.938	.516	238
240	397.4	371.0	829.8	1200.8	1.923	.520	240
242	398.2	371.7	829.2	1200.9	1.907	.524	242
244	398.9	372.5	828.5	1201.0	1.892	.528	244
246	399.6	373.3	827.8	1201.1	1.877	.532	246
248	400.3	374.0	827.2	1201.2	1.862	.537	248
250	401.1	374.7	826.6	1201.3	1.848	.541	250
275	409.6	383.7	819.0	1202.7	1.684	.594	275
300	417.5	392.0	811.8	1203.8	1.547	.647	300
350	431.9	407.4	798.5	1205.9	1.330	.750	350

TABLE II

Properties of Common Substances

	Specific Gravity.	Weight per Cubic Foot, Lbs.	Weight of One Cubic Inch, Lb.	Specific Heat.	Coefficient of Expansion per Deg. F.	
					Linear.	Volumetric.
Aluminum	2.60	161.	.095	.212	.000011	.000033
Bismuth	9.82	613.	.353	.031	.000008	.000024
Brass	8.10	503.	.293	.094	.00001	.00003
Copper	8.79	545.	.318	.092	.000009	.000028
Coal (anthracite)	1.43	88.7	.058	.241		
Coke	1.00	62.4	.037	.203		
Gasoline	.68	42.4				
Glass	2.89	180.7	.105	.198	.000005	.000014
Gold	19.26	1200.	.697	.032	.000008	.000024
Ice (at 32° F.)	.92	57.5	.033	.504		
Iron (cast)	7.5	465.	.271	.130	.000006	.000018
Iron (wrought)	7.74	582.	.280	.114	.000007	.000021
Lead	11.35	708.	.411	.031	.000016	.000048
Limestone	3.16	197.	.114	.217		
Mercury (at 32° F.)	13.60	849.	.492	.033	.000033	.000100
Cement	2.24	140.	.083	.20	.000008	.000024
Nickel	8.90	547.	.321	.109	.000007	.000020
Platinum	21.5	1342.	.779	.032	.000005	.000015
Pine (white)	.55	34.	.020	.65	.0000025	.000008
Silver	10.47	653.	.379	.056	.000011	.000033
Steel	7.83	486.	.292	.116	.000007	.000020
Tin	7.29	452.	.264	.056	.000012	.000035
Zinc	7.19	445.	.260	.095	.000016	.000048

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